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Electromechanical transmission challenges in non-road mobile machinery drivetrains

Thesis submitted as partial fulfilment of the requirements for the degree of Masters in Science (Technology)

Espoo, 26 November 2014

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Abstract

Increasing energy costs are forcing non-road mobile machinery (NRMM) manufacturers to produce more efficient drivetrains. Developing hybridization of drivetrains is facing many challenges. One typical situation is when electromechanical power transmission challenges arise when adding an electric motor to an otherwise conventional type of drivetrain. The main engineering targets are the great differences in torque and speed characteristics for electric motors vs. combustion engines.

The aim of this thesis is to find different alternatives of down gearing an electric motor and to explain the characteristics of them. The purpose of this thesis is also to serve as a concept level design guide for those who want to replace the combustion engine with an electric motor and need to down gear the motor especially in NRMM purposes. This thesis introduces the reader to the most typical hybrid and electric drivetrains and how the electromechanical transmission challenges appear in the different types of drivetrains. The main differences in car and NRMM drivetrains are introduced as well as the conventional drivetrain briefly. Furthermore the design perspective of a hybrid drivetrain is introduced. Different subsystems and typical matters to take into account when designing a hybrid drivetrain are discussed. In addition, characteristics of different types of power transmission are introduced to the reader. Also comparison of the different power transmission types is performed to help the reader in a hybrid drivetrain concept design phase. This comparison is focused on the torque and rotational speed problematics.

The main result of the thesis is the case study. The case study is presented in the end of the thesis. This case study was performed in Aalto University at the Vehicle Engineering research group for the Tubridi-project. In this case study two electromechanical transmissions for NRMM were designed and parts ordered. Belt drive transmissions were selected to be designed within this thesis work. Gear ratios resulted as 3:1 for the belt drive on the front axle drivetrain and 2:1 for the rear. Belt drive parts were ordered from two different manufacturers. NRMM in focus was a 14 ton underground mining loader. Mechanical design drawings are presented in the end of the thesis as attachment. A belt drive turned out to be possible to implement to the target work machine.

Keywords belt drive, hybrid, electric, drivetrain, transmission, non-road mobile machinery, underground mining loader, efficiency, gear

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Tiivistelmä

Työkoneiden energiankulutusta pyritään pienentämään parantamalla ajovoimalinjojen hyötysuhdetta. Eräs keino hyötysuhteen nostamiseen on ajovoimalinjojen sähköistys ja hybridisointi. Sähkö- ja polttomoottorilla on eri vääntökäyttäytyminen, mikä aiheuttaa haasteita liitettäessä sähkömoottori perinteiseen ajovoimalinjaan.

Tämän työn tavoitteena on löytää erilaisia sähkömoottoriin liitettäviä voimansiirtoratkaisuja, joilla voidaan alentaa pyörimisnopeutta. Eri voimansiirtoratkaisujen vääntömomentteja, pyörimisnopeuksia ja erityispiirteitä arvioidaan ja vertaillaan hybridiajovoimalinjan konseptisuunnittelun tueksi. Työn tarkoituksena on myös toimia konseptitason suunnittelun apuvälineenä varsinkin tilanteessa, jossa työkoneen polttomoottori halutaan korvata sähkömoottorilla ja tarvitaan alennusvaihde. Työssä esitetään tyypillisimmät hybridi- ja sähköajovoimalinjat ja miten sähkömekaanisen voimansiirron haasteet ilmenevät niissä. Olennaiset erot autojen ja työkoneiden ajovoimalinjojen välillä esitetään. Hybridiajovoimalinjojen suunnitteluperiaatteet esitetään.

Työn merkittävin tulos on käytännön osuus, joka suoritettiin Aalto-yliopistossa Ajoneuvotekniikan tutkimusryhmässä Tubridi-projektille. Tässä käytännön osuudessa suunniteltiin ja tilattiin osat kahteen sähkömekaaniseen voimansiirtoon 14 tonnin kaivoslastaajaa varten. Hihnaveto valittiin tässä työssä suunniteltaviksi voimansiirroiksi. Etuakselin hihnavedon välityssuhteeksi saatiin 3:1 ja taka-akselin välityssuhteeksi 2:1. Hihnavetojen osat tilattiin kahdelta eri valmistajilta. Osien ja kokoonpanojen työkuvat on esitetty työn lopussa. Hihnaveto osoittautui mahdolliseksi toteuttaa voimanvälitystavaksi kohteena olleeseen työkoneeseen.

Avainsanat hihnaveto, hybridi, sähkö, ajovoimalinja, voimansiirto, työkone, kaivoslastaaja, hyötysuhde, vaihde

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At this very moment I would also like to thank my fellow students and all the vehicle engineering people for their support during this thesis work. Many interesting conversations are left in the memory from these times.

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Espoo 26.11.2014

Jan Liljeström

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Abbreviations

BMS	Battery Management System
CVT	Continuously Variable Transmission
ECU	Engine Control Unit
LHD	Load-Haul-Dump
NRMM	Non-Road Mobile Machine
SOC	State of Charge

Terms

A	Cross-sectional area
C_D	Drag coefficient
F_i	Drag force
F_n	Hill climbing resistance
F_R	Total driving resistance force
F_r	Rolling resistance force
f_r	Rolling resistance coefficient
F_x	Tire peripheral force
F_Z	Wheel load
i_{bd}	Belt drive gear ratio
i_d	Differential gear ratio
i_{hg}	Gear ratio of hub gear
i_t	Transmission gear ratio
i_{tot}	Total gear ratio of drivetrain
M_m	Electric motor torque
η	Drivetrain efficiency
ρ	Air density
r_k	Tire radius as loaded
v_x	Velocity

1 Introduction

The rising cost of energy and the increasing emission regulations force non-road mobile machinery (NRMM) manufacturers to develop more efficient drivetrains. There is also a continuous search for more effective and powerful machines to achieve more tons per hour. Also more accurate control systems, lower maintenance costs, and technology that achieves a higher level of automation is in focus. All these matters cause the manufacturers to start thinking of alternative ways of producing energy other than by burning petroleum or diesel. This is where electricity is making its breakthrough. Through electrification and hybridization, different business opportunities in service arise via new rental models and leasing operations.

Electricity has been growing as an energy form. There are a number of ways to generate electricity without having to put too high an impact on the environment. This generated electric energy can be transferred a long distance - all the way to a battery charger and thereby to an onboard battery on NRMM. The transmission of electric energy from the source all the way to the battery is more efficient than when producing and delivering petroleum or diesel. In figure 1 the well to tank efficiency as well as the tank to wheel efficiency is presented for both an electric passenger car as well as a petroleum passenger car. In this case the well, the source of the energy is the same. Electricity can also be generated from energy sources much more efficient than this. This means that the generation efficiency seen in figure 1 as 33% is for converting hydro energy into electricity up to 95% in case of a large hydro power plant (EURELECTRIC, 2003).

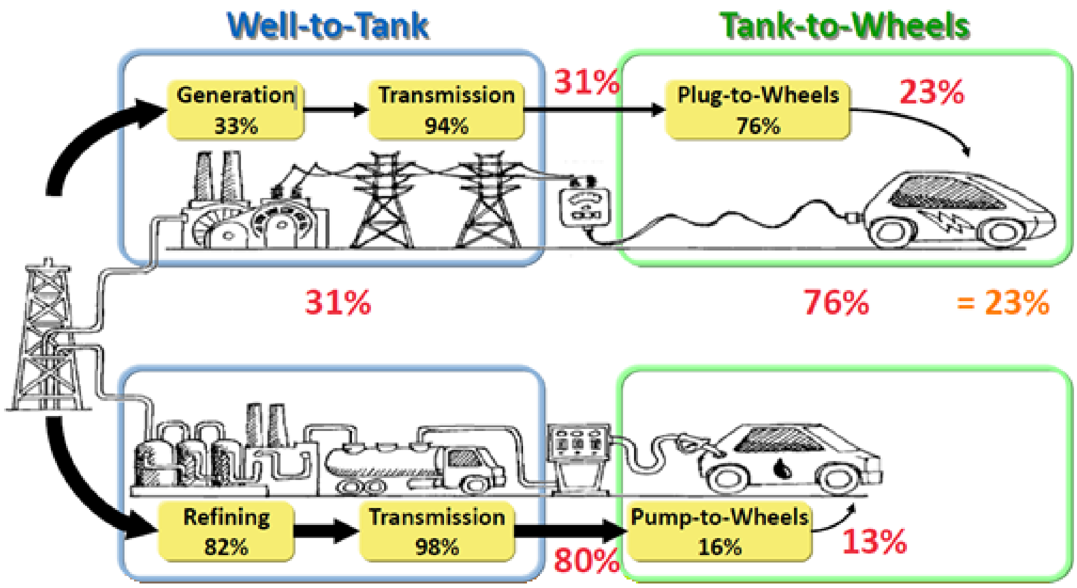


Figure 1. Well to wheel efficiency for electric and petroleum car. (Nesea, 2012)

The transition from NRMM with only an internal combustion engine traction force to NRMM with only electric motors and batteries providing traction force is in progress. However, this transition does not happen in one single step. There will be a long period of transition where different types of energy source combinations will be tried. During this time, issues with matching different kinds of power sources will be encountered. Components belonging to different kind of power sources such as electric motors will replace the internal combustion engine as a traction motor and the internal combustion engine will be used instead as a power source to generate electric energy. This replacement will cause challenges for many drivetrain designers since the revolutions per minute will increase significantly for the first steps of the drivetrain. This is because the typical speeds for electric motors are many times greater than those for internal combustion engines. For the end of the drivetrain to be able to still operate in the same way as before, there arises a need of down gearing the electric motor. The aim of this thesis is to find different alternatives of down gearing an electric motor and to explain the characteristics of them. The purpose of this thesis is also to serve as a concept level design guide for those who want to replace the combustion engine with an electric motor and need to down gear the motor especially in NRMM purposes.

In this thesis the traditional drivetrains as well as the differences in characteristics for passenger vehicles and NRMM are introduced. Also hybrid and electric drivetrains are discussed shortly. After this, the thesis concentrates on the challenges with electric transmission in NRMM drivetrain design. Different types of power transmissions are introduced and compared. The main output is a design case included in this thesis performed on certain NRMM.

In this thesis the word “drivetrain” is used for the mechanical transmission components beginning from the source that provides the mechanical rotational power to the wheels and including all the components all the way to the wheels. The wheel itself is not considered in this study.

The words “electromechanical transmission” in this thesis stands for the parts of a drivetrain where electric energy is converted to mechanical rotational movement. This term also includes the parts of the drivetrain where this conversion has a great impact considering the significant differences in torque and speed compared to conventional drivetrains.

1.1 Traditional drivetrains

The main purpose of the drivetrain is to provide the driving wheels with enough power and torque that is greater than the drag forces that affect the NRMM at the moment. The combustion engine in the drivetrain is the component which converts the chemical energy into mechanical rotational motion. These combustion engines are typically named otto or diesel engines based on their burned fuel or combustion process ignition moments. In traditional drivetrains with internal combustion engines the engine always has an idling speed and therefore it operates between the idling speed and the maximum speed in normal conditions. In some cases the engine runs at a constant speed. The maximum torque and power is not achieved at the same rotational speeds. In practice, below the idle speed there is no torque to be exploited from the internal combustion engine. This means that there has to be a component in the drivetrain that is in theory a variable transmission having a gear ratio from infinite to 1 - i.e. a clutch. The purpose of the different transmissions in the drivetrain is to convert the torque and speed of the motor to the appropriate needed speed and torque at the wheels (Dietsche, Crepin, & Dinkler, 2002).

A traditional drivetrain typically consists, in simplified version, of internal combustion engine, clutch, transmission and drive shafts to wheels. The mechanical power produced by the engine is most commonly transferred all the way to the wheels via mechanical rotating parts. The mechanical energy from the motor can also be transferred to the wheels in several other different ways. One example is using hydraulic components. A typical drivetrain for passenger cars can be seen in figure 2.

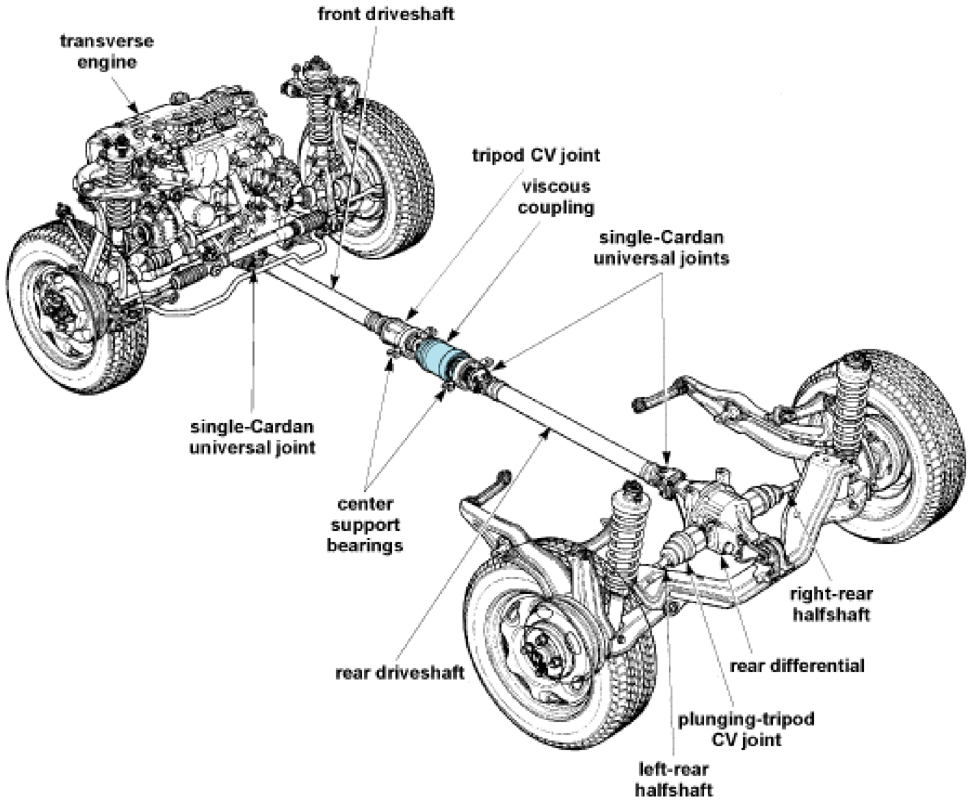


Figure 2. Typical drivetrain for a rear wheel driven passenger car. (Abhi, 2012)

The engine in a traditional drivetrain is the least efficient single component, having efficiency typically below 40% (Takaishi, 2008). In practice the efficiencies are usually even lower than this. In figure 3 thermal efficiencies for different kind of combustion engines are presented. A great percentage of the wasted energy is transferred to heat. Especially in cold climates this heat can be used for heating the cabin and the overall efficiency is increased that way. All the other rotating parts through the drivetrain have losses in the form of friction which also produces heat. However, this heat is not beneficial.

The transmission in a traditional drivetrain is typically equipped with several gears with gear ratios generally ranging from 1 to 16 (Dietsche et al., 2002). In passenger cars the gear ratios are typically below 5.5 and in some commercial vehicles the ratio can be as high as 16 (Dietsche et al., 2002). The most common transmission type is the gear wheel type gearbox. In addition to the gearbox, transmission gear ratios can also be found in several other points of the drivetrain, such as the differential and the hub gear.

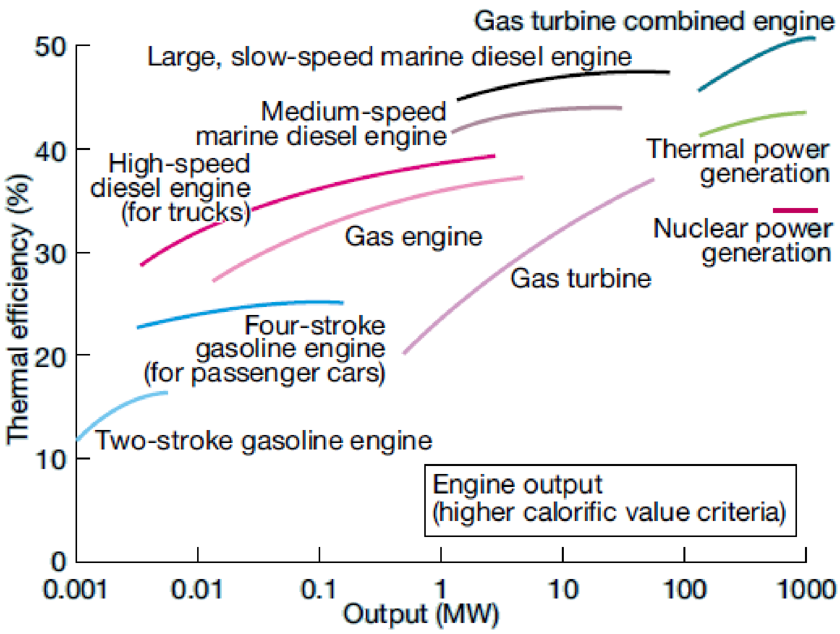


Figure 3. Thermal efficiencies of various types of small- to medium-sized diesel and gas engines. (Takaishi, 2008)

1.2 Differences in characteristics for cars and non-road mobile machinery drivetrains

The main differences in characteristics for cars and NRMM are because the main purpose they are made for is different. Cars are mostly for individual use, travelling from one place to another. A car is usually much lighter than NRMM and this causes the forces that are transmitted through the drivetrain to be of a smaller magnitude. When a common purpose for cars is travelling, the drivetrain in a car has to work for much higher driving speeds than for general NRMM. The NRMM has to work many times in noticeable demanding conditions having significant variation in payload which put an additional demand on the drivetrain. A car operates mostly on the road with small or even minor variation in payload. Typically in cars there might be only one to two passengers.

$$P = T * \omega \tag{1}$$

As seen in equation (1), the power transmitted through the drivetrain is equivalent to the torque and speed. The forces in the drivetrain are also equivalent to the torque. Since the power is greater and the speed is less for NRMM, the torque i.e. the forces in the drivetrain are of a much greater value than for cars. This is the major difference between cars and NRMM. For this reason NRMM drivetrains are more demanding to design and manufacture. When this needs to fit in small space it is more expensive.

By definition (EU, 1997), non-road mobile machinery (NRMM) shall mean any mobile machine, transportable industrial equipment or vehicle with or without body work, not intended for the use of passenger- or goods-transport on the road, in which an internal combustion engine having a net power that is higher than 18kW but not more than 560 kW is installed and operates under intermittent speed rather than a constant speed. The term NRMM covers, for example, construction, agricultural, forestry and snow-plough equipment.

The drivetrain components in a car are typically engine, clutch, gearbox and drive shafts to wheels. In NRMM there can be in addition to the drivetrain components for a car many additional components that are needed to achieve the overall needed gear ratio for the drivetrain.

For NRMM terramechanics play a greater role than for cars. Terramechanics stands for the interaction of wheeled or tracked vehicles and various surfaces. This terramechanic comes in to play because NRMM usually operate on unprepared ground and need to overcome complex and difficult ground obstacles, such as steep grade and soft ground (Ehsani, Gao, & Emadi, 2010). Motion resistance caused by terrain compaction and bulldozing is added to the total running resistance for the NRMM. In underground mining the ultimate obstacle is considered to be a piece of rock that has dropped from the ceiling and ends in front of the tire of the NRMM. The terramechanics puts an additional demand on the traction force of the NRMM. The great need of traction force on NRMM is commonly not solved by the motor or engine sizing but instead with the transmission gear ratio and driving wheel radius. This is greatly different to passenger cars where the gear ratios have to be smaller to achieve higher top speed in only a few gears. Furthermore, NRMM have, in most cases, many times greater payload than a passenger car. Starting from zero speed is more demanding for NRMM. In passenger cars this can be solved by the clutch, but with high payload vehicles there needs to be a reduction gear.

There are some major differences in operating NRMM to cars that are worth considering. For NRMM there is plenty of back and forth movement possible with symmetric velocities; in contrast cars seldom reverse and have only one slow reversing gear. NRMM drive cycles typically include more starts and stops than passenger car drive cycles. Changing gear when driving can cause a challenge in some situations for NRMM especially in case of soft ground or obstacles in the way. For passenger cars, changing gear can sometimes even be considered to cause a “cool” effect. For safety reasons, brakes in passenger cars versus NRMM are typically implemented in a different way. For NRMM brakes are integrated to the drivetrain and are of a releasing type i.e. the brakes are released when braking instead of having an adjustable force applied to the brakes as in passenger cars. This means that in case something goes wrong the spring loaded brakes in NRMM are applied. For passenger cars this type of brake would be too dangerous at high speeds. For some NRMM brakes are not even needed when the high gear ratios added to the inertia of the drivetrain causes certain NRMM to reduce their velocity rapidly when tractions force is cut.

2 Hybrid- and electric drivetrains

Hybrid drivetrain stands for a drivetrain where two or several power sources are combined in order to eliminate or reduce the disadvantages of each and combine the advantages. In this thesis, all the hybrid drivetrains considered are drivetrains where one of the power sources in the combination is electric. This is most common for hybrid drivetrains. In this chapter also pure electric and diesel electric drivetrains are considered.

Electric drivetrains have advantages such as less noise, exhaustlessness and good efficiency. According to (Dietsche et al., 2002), as opposed to drivetrains with an internal combustion engine, the power output of an electric drivetrain is most commonly limited by the power of the energy storage. Because of this the electric motors need to be designed or chosen to work for the battery characteristics. The energy storage needed for an electric motor is currently also limiting the operation time of the NRMM much more than an energy storage for an internal combustion engine. Therefore, it is a great advantage to build a hybrid drivetrain instead of a fully electric one. In a hybrid drivetrain the electric motor is used whenever possible. Typically this is limited by the charging of batteries. Depending on the type of hybrid system and the topology of the drivetrain, the energy storage for the electric motor is charged when the car is plugged to a cord or by the combustion engine or regenerative braking while driving. It has also been claimed that hybridization is a way to handle the riskiness of battery technology especially in terms of durability and performance. In table 1 we can see that a plug-in hybrid system is the only one that needs charging at a power socket. Other types of hybrids produce the electricity onboard.

Table 1. Functions and hybrid systems. (Reif, Dietsche, & Et.al, 2011)

		Hybrid systems			
		Start/stop	Mild hybrid	Strong hybrid	Plug-in hybrid
Function	Start/stop	●	●	●	●
	Regeneration	●	●	●	●
	Electric assistance		●	●	●
	Electric driving			●	●
	Charging at power socket				●

As seen in table 1, hybrid vehicles can be classified on the basis of their functions. As we can see in the table, all of the hybrid systems include the start/stop function as do electric drivetrains. This feature allows the vehicle to shut down when the engine or motor is not needed. This results in a higher efficiency of the drivetrain over conventional vehicles with combustion engine, since the combustion engine has to keep running on idle speed to keep the auxiliary devices running, which consumes a lot of energy. In hybrid and electric vehicles, the auxiliary devices are powered by the batteries. A great amount of energy is consumed for a vehicle during acceleration when adding kinetic energy to the vehicle. When braking, in conventional vehicles this kinetic energy is turned to heat, which is a great loss. Regenerative braking in hybrid and electric vehicles indicates that part of their kinetic energy is transformed to electric energy charged to the battery by braking the vehicle partly with a generator instead of only with the conventional friction brakes. According to (Reif et al., 2011) with a start/stop systems fuel savings up to 5% can be achieved; with a mild hybrid up to 15%; with a strong hybrid up to 30%; and with a plug-in hybrid up to 70%. According to (Vauhkonen et al., 2014) with an electric drivetrain, fuel savings up to 80% can be achieved. In case of regenerative braking, it might also be worthwhile to consider how well the control is performed via mechanical brakes as compared to controlling the deceleration or maintaining a constant velocity by traction control.

A great advantage for the hybrid drivetrain compared to a fully electric one is that the operation time can be extended enormously. This might cause limitations in terms of performance, but the work process will still continue. Many types of hybrid systems don´t even need to be plugged to a cord in order to continue to work as hybrid. These types of hybrid systems just need to have the fuel tank refilled and the operation time becomes unlimited.

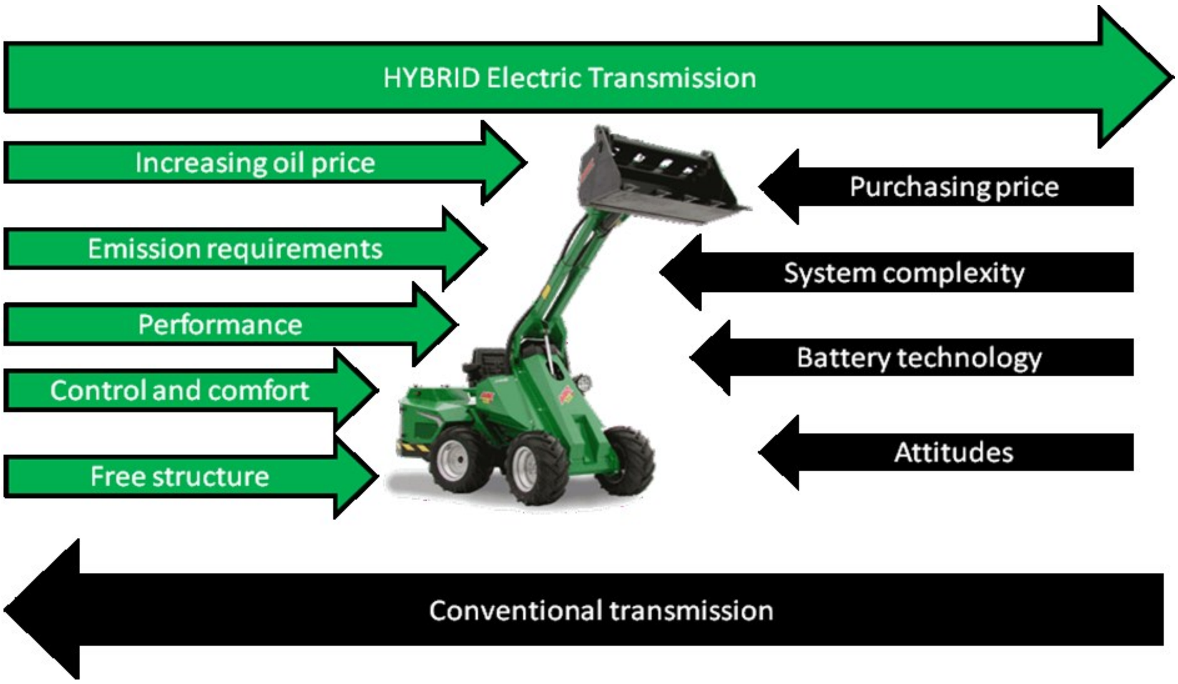


Figure 4. Drivers towards Hybrid Electric transmission. (Suomela, Lehmuspelto, & Sainio, 2010)

As we can see in figure 4 there is a competition between hybrid electric transmissions and conventional transmissions. There are cases where the competition is much harder for the conventional transmission than for hybrid electric or fully electric transmission. For example, when having to work in enclosed sites, the emissions cause difficulties. In mines the ventilation cost for getting the exhaust gases and heat out is almost three times larger than the price of the fuel burned (Vauhkonen et al., 2014). For some indoor working it might even get too expensive to arrange ventilation. Therefore, fully electric transmission is truly an advantage.

The emission regulations are tightening all the time as we can see from figure 5. In the figure we can also see that emission regulations are different for engines in different power ranges. In conventional drivetrains the combustion engine needs to be designed to cover the peak power need of the machine. A great advantage with hybridization is that the combustion engine can be downsized since the peak power need of the machine can be covered with the electric motor. This way, there is a possibility for the downsized engine to end up in a lower power range with looser emission regulations. This we can see in figure 5. We can also see that the lowest power range does not have any emission regulations at all - at least before year 2017. Speculations have been made about having several engines with low power output instead of one with higher power output to avoid emission regulations. It is also worth considering internal rules of global companies. Even though in some parts of the world the emission regulations are not as demanding as for other parts of the world, companies still like to maintain the same standard for their product throughout the world. This way a global company can promote their environmental and worker protection philosophies.

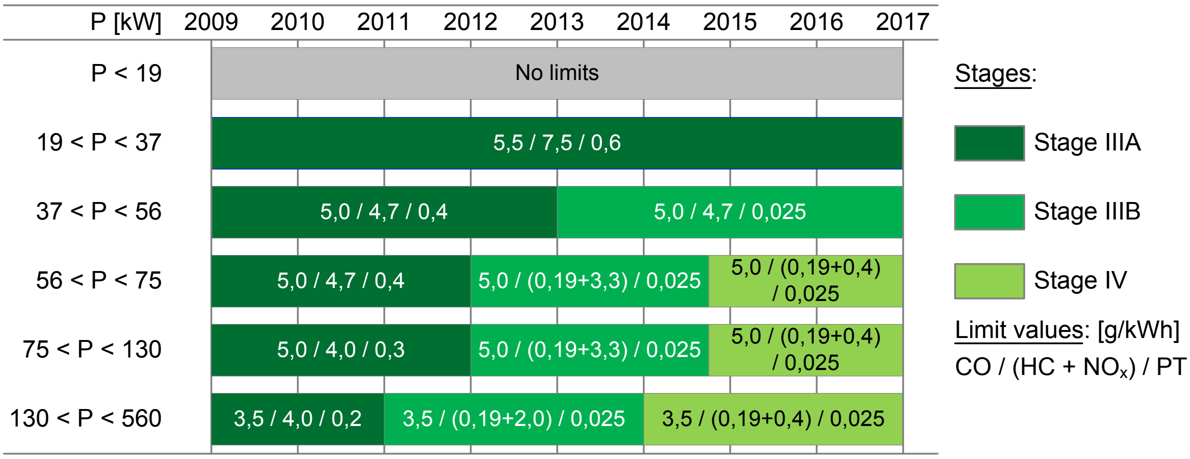


Figure 5. European Union regulations for emission standards considering diesel engines other than constant-speed engines to be installed in non-road mobile machinery. (Suomela et al., 2010)

2.1 Drivetrain architectures and examples

In order to give a small introduction to hybrid and electric drivetrains, in this chapter the most common architectures will be introduced as well as one example of each drivetrain architecture. All of the architectures are not yet widely used in NRMM drivetrains, so some of the examples are prototypes not commercially available.

2.1.1 Electric drivetrain

The first drivetrain to be introduced is the fully electric drivetrain. This is the simplest of the drivetrains considered in this thesis. All the other drivetrains except the diesel-electric, include all the components that are in the electric drivetrain. In an electric drivetrain, as well as in all the other different types of drivetrains considered in this thesis, there are electromechanical transmission challenges that are the key point of this thesis. The main challenge is that the electric motor included in all these drivetrains has different torque and speed characteristics than a conventional combustion engine. This puts an additional demand on the mechanical drivetrain components next to the electric motor. Although all these different drivetrains have this challenge, it appears in different ways for the different types of drivetrains. In an electric drivetrain only the electric motor is mechanically connected to the driving wheels. Since an electric motor typically turns 2-4 times faster than a combustions engine, there is need for a transmission between the wheels and the motor that can handle greater differences in rotational speeds. Different solutions for this challenge are discussed further in chapter 4.

An electric drivetrain consists of batteries and an electric motor as seen in figure 6. In an electric drivetrain, some components such as regenerative brakes seen in figure 6 generate electricity from kinetic or potential energy and charge the batteries with it. In conventional drivetrains, this kinetic energy is turned to heat or affected by a power source that needs additional energy.

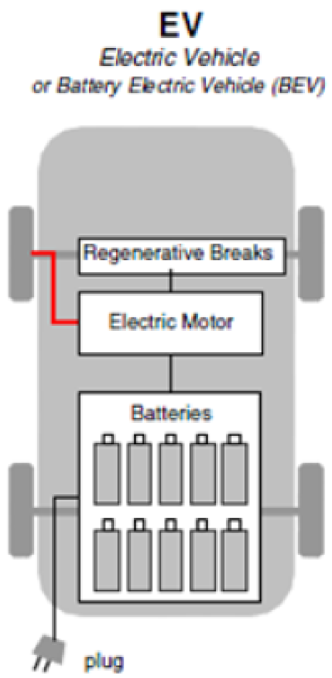


Figure 6. Generic Battery powered Electric Vehicle architecture. (Chih-ming & Jheng-cin, 2010)

As previously stated, the fully electric drivetrain has great advantages in some fields that other drivetrains can´t compete with. For example, more than 50% of the indoor use of NRMM is electric because there are no emissions (Dietsche et al., 2002). Noiselessness is also hard to achieve with drivetrains other than electric ones. However, there are still many challenges with electric vehicles and battery electric vehicles. The battery technology is the most limiting factor in battery electric vehicle success. Batteries are still expensive. Because the energy density is much lower than for fuel, there is a need for huge batteries with a great mass in case the vehicle would be wanted to compete with conventional vehicles in operation time or range. The power grid of work sites would also need to be upgraded in order to be able to perform fast charging of batteries. In case electric vehicles or NRMM do not have a battery, they need to be connected to the grid all the time when the machine operates. This causes limitations in the machine mobility as well as a mess of wires here and there on the work site. There are a number of this type of electric NRMM but they mostly perform stationary work or need simple mobility. There are electric large sized underground mining loaders which mostly drive back and forth and do loading and unloading. These are also typically electric with an electric wire on a reel in the back of the loader.

Figure 7. Electric excavator. (Liljeström & Isomaa, 2014)

In figure 7 is seen an electric excavator that was built as a student project in Aalto University during the academic year 2013-2014. This excavator formerly had a diesel engine powering the hydraulics. By only changing the power source to an electric motor, energy savings of about 80% were achieved (Liljeström & Isomaa, 2014). This gives us an idea of the differences in efficiency between conventional combustion engines and modern electric motors. In this excavator case, more energy saving would have easily been achieved by adding features like energy regeneration from lowering bucket or turning cabin, start and stop function or replacing conventional hydraulics with direct driven hydraulics. The aim of the study was mostly to show how easily NRMM can be turned to electric and how well it works. However, in the excavator case batteries are not always necessary due to the rather stationary work. Instead, provision of electricity could be implemented by connecting to the grid. Leaving the batteries out of the design would make the excavator significantly cheaper to manufacture.

There are a growing number of companies offering parts for electric drivetrains these days. As an example in the electric excavator case, the batteries are Altairnano battery modules made in USA (Altairnano, 2014), which offer relatively high power. Another similar battery manufacturer is Kokam in Korea (Kokam, 2014). The electric motor in the excavator is of an inexpensive type made by Golden Motor in China (Golden Motor, 2014). The motor controller is produced by Sevcon in UK (Sevcon, 2014).

2.1.2 Diesel-electric drivetrain

A diesel-electric drivetrain is not a hybrid drivetrain but can easily be mixed with hybrids. The easiest way to recognize the difference between a diesel-electric and a hybrid drivetrain is to see if there is an energy storage system between the generator and the electric motor or not. This is the main difference and excludes the possibility of achieving other features known for hybrid drivetrains. The principal layout of a diesel-electric drivetrain is similar to the one presented for series hybrid in figure 9 except that there is no energy storage system so the electricity is connected from the generator straight to the electric motor.

A diesel-electric drivetrain in NRMM has a great advantage compared to conventional diesel-mechanical drivetrain since there is no need of a mechanical clutch. Diesel-electric drivetrains are typically implemented in heavy NRMM such as locomotives, ships and large trucks. In figure 8 can be seen the diesel-electric drivetrain of a large mining truck. For this type of NRMM which carry a huge load, the diesel-electric drivetrain can provide maximum torque at the lowest rotational speeds when the machine is about to start. Electric motors are also able to start from zero speed in opposition to diesel engines, which need a clutch. The torque puts a high demand on mechanical clutches in this type of heavy NRMM and there is a greater risk of clutch breakdowns. This challenge is beaten by the diesel-electric drivetrain. In ships, a diesel-electric drivetrain gives much better maneuverability with technologies such as ABB Azipod (ABB Azipod, 2014).

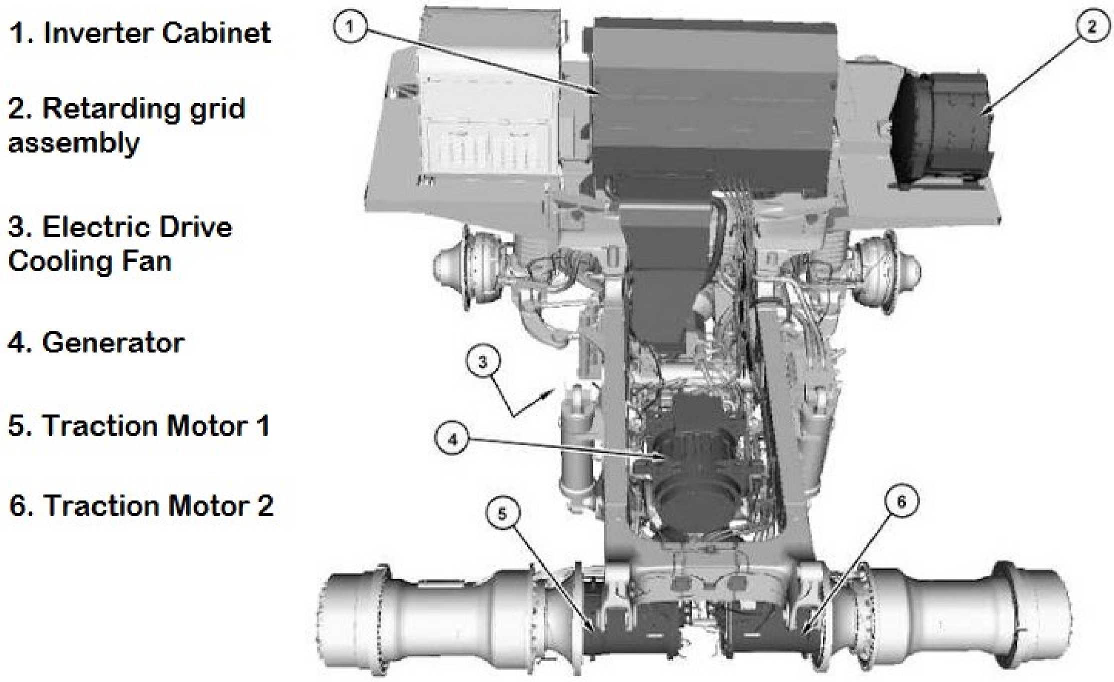


Figure 8. Diesel-electric drivetrain of a large mining truck. (Manolo, 2013)

The challenges for the electromechanical transmission in diesel-electric drivetrains is the main focus of this thesis. Even though the case study NRMM is not diesel-electric, but series hybrid, they both use the same type of components between the traction motors and the driving wheels. The challenge occurs when there is a need of high torque and light weight or compact shape of the motor. A characteristic for electric motors is that the greater the face width of the electric motor the greater the torque. This means that to achieve high torque with a small electric motor, it needs down gearing. Different solutions for this down gearing are discussed further in chapter 4.

There are manufacturers of parts for heavier work machines such as electric motor and generator manufacturers ABB Motors and Generators in Finland (ABB Motors and Generators, 2014) and Siemens in Germany (Siemens, 2014). Both of the companies also provide motor controllers.

2.1.3 Series hybrid drivetrain

According to (Reif et al., 2011) a series hybrid drivetrain is defined to have a series connection from the perspective of the energy flows. Series hybrids are always strong hybrids (Reif et al., 2011), see table 1. Series hybrid drivetrains are similar to diesel-electric drivetrains as mentioned earlier in chapter 2.1.2 but also have many similarities to the electric drivetrain introduced in chapter 2.1.1. The main difference to the diesel-electric drivetrain is that a series hybrid drivetrain includes energy storage system between the generator and the electric motor as can be seen in the principal layout for series hybrid drivetrains in figure 9. This energy storage works like a buffer in between the energy generator and consumer. This buffer allows the engine which is generating the electricity to turn at a constant speed so that the power of the generator is equal to the average power consumption. This way the engine can run constantly at a good efficiency. As opposed to diesel-electric drivetrains and similar to electric drivetrains, in series hybrid drivetrains energy can be regenerated to the energy storage system.

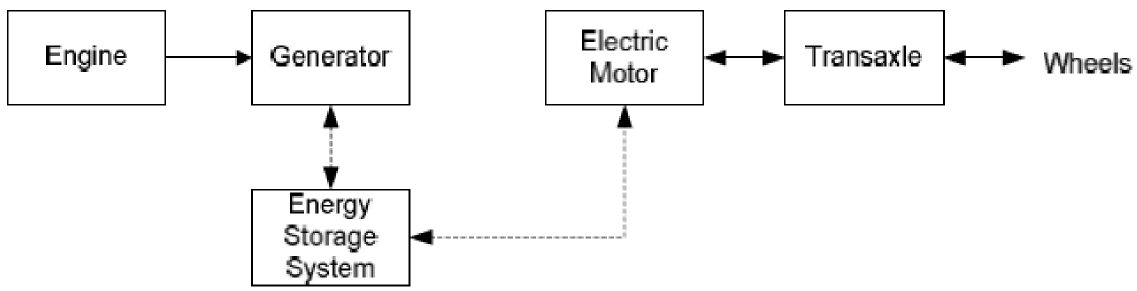


Figure 9. Generic Series Hybrid Vehicle architecture. (Zhou, 2011)

Different to electric drivetrains, the series hybrid has at least one additional energy storage except the one seen in figure 9. This energy storage is most commonly a fuel tank. The fuel tank is easily refilled and the range is extended. There are some cars that are purely electric but most are of this serial hybrid type as electric vehicles with “range extender”. The same trend is seen for outdoor use NRMM. This is because the operation range provided by batteries is in many cases not enough, so purely battery electric NRMM are not in high demand on the market at the moment. When the battery technology develops to produce cheaper, more reliable batteries with a higher energy density, the purely electric drivetrain will be more popular. Series hybrid is a transition phase drivetrain between the conventional and the purely electric drivetrain.

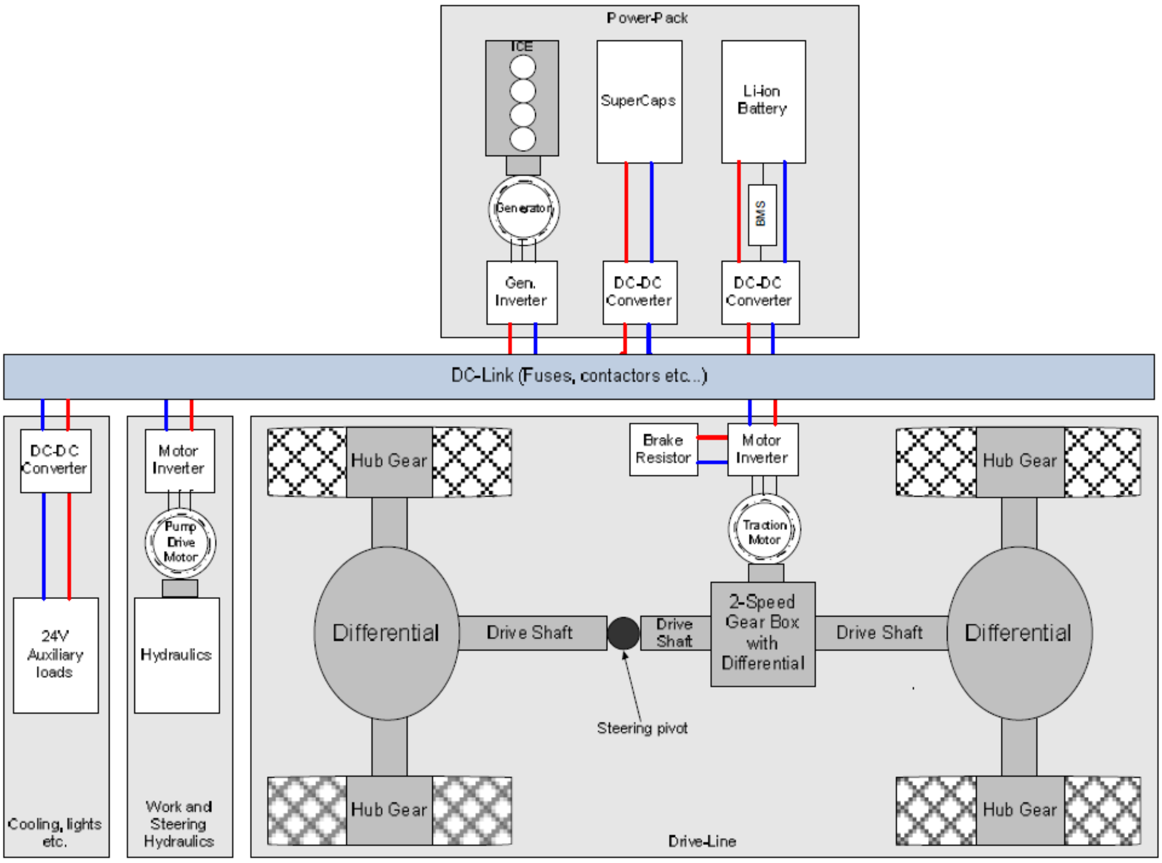


Figure 10. Concept of Series Hybrid mining loader. (Lehmuspelto, Heiska, Leivo, & Hentunen, 2009)

Figure 10 represents a series hybrid NRMM drivetrain layout of an underground mining loader. This is an earlier stage design of the same underground mining loader which is discussed further in this thesis chapter 5. This example shows that in a hybrid drivetrain there can be more energy sources than just two. The advantages of adding super capacitors in a hybrid drivetrain are that they are able to be charged and discharged at incredible speed. This allows regeneration of peak braking powers and covering of peak power needs. This earlier stage drivetrain layout was later changed so that instead of having a common traction motor for both wheels, there are now two separate traction motors for front and rear axles. This overcomes the physical challenges of having a drive shaft through the long machine, including steering pivot. Separate motors give better controllability as advantages. However, there is not the possibility to direct all traction force to one single tire, as there is in case of one common motor. The high need of traction force and the extremely limited space in this mining loader puts a huge demand on the transmission next to the electric motors. The design and development of this transmission is considered further in chapter 5.2.

2.1.4 Parallel hybrid drivetrain

According to (Reif et al., 2011) a parallel hybrid drivetrain is defined to have two or more parallel energy flows that are added up to the total drive power. Parallel hybrids can be either mild or strong hybrids (Reif et al., 2011), see table 1. The principal layout of a series hybrid drivetrain can be seen in figure 11. The parallel hybrid drivetrain is different from series hybrid, as can be seen from the figure. A major difference is that there is a mechanical connection between engine and wheel as opposed to series hybrid drivetrains. This is an additional challenge for the transmission system compared to the series hybrid, since here the rotational speed of the electric motor does not only need to be fit to the rotational speed of the wheels, but also to the rotational speed of the combustion engine. This causes the transmission part seen in figure 11 to be more complicated.

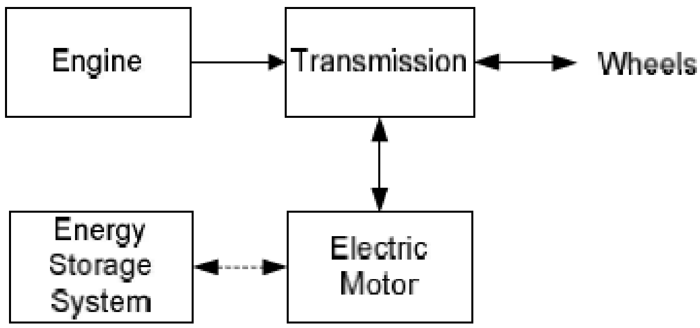


Figure 11. Generic Parallel Hybrid Vehicle Architecture. (Zhou, 2011)

An advantage of a parallel hybrid is that there is no back and forth converting of mechanical energy to electric energy, as in series hybrids. This aids the drivetrain to achieve better efficiency especially under high load operation. The conventional drivetrain can be operated over a wide range of speed and torque. This is a fundamental advantage of the parallel hybrid. The parallel hybrid drivetrain is the hybrid drivetrain which is the least different from the conventional drivetrain and therefore it is the easiest hybrid drivetrain to be converted from conventional drivetrain. Since the electric part of the drivetrain does not need to be of such a high power as for series hybrid, this results in an even cheaper conversion. This is why parallel hybrid drivetrains are more popular than series hybrid drivetrains.

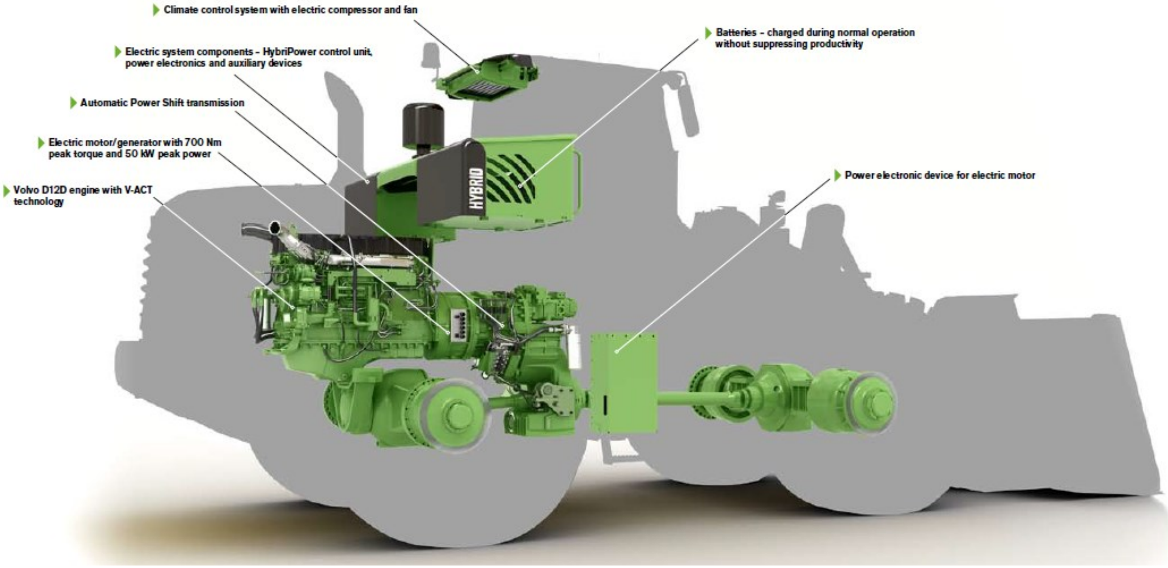


Figure 12. Volvo wheel loader L220F Hybrid drivetrain. (Volvo, 2008)

In figure 11 there is presented a Volvo wheel loader L220F Hybrid drivetrain as an example of a parallel hybrid drivetrain in NRMM. This drivetrain is especially designed for the wheel loader application where there is a great amount of idle time for the machine present. This idling time comes from the waiting on the next truck when the loader has already loaded the previous one or other similar work site related matters. With a hybrid drivetrain the fuel consumption caused by a combustion engine during idling can be removed. This is made by the so called "Start/stop" –function, see table 1. According to (Volvo, 2008) this hybrid conversion of the wheel loader increases productivity while it at the same time saves fuel up to 10%. It is worthwhile to consider that this type of layout does not need changes in axle construction. Axles including brakes and suspension are expensive to re-engineer. Therefore a cost effective solution is achieved when adding only an electric motor after the combustion engine to gain the benefits of a hybrid drivetrain without a need to re-engineer major components.

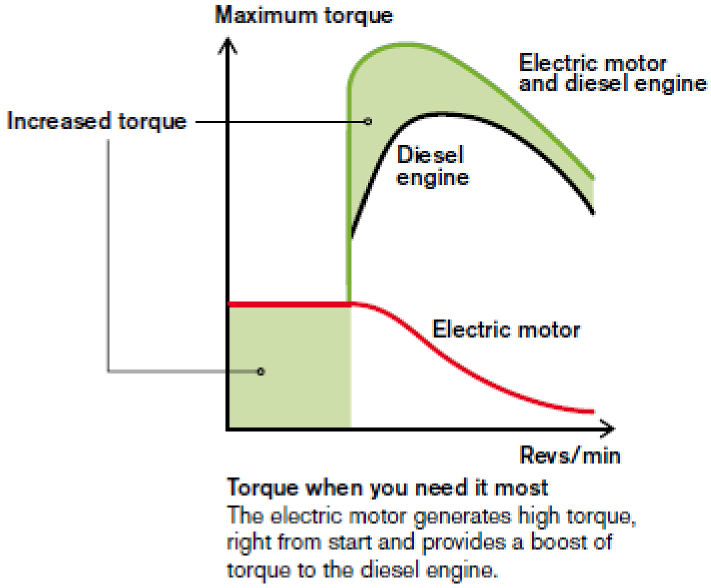


Figure 13. Motor and engine characteristics for Volvo wheel loader. (Volvo, 2008)

The advantage in torque for a parallel hybrid drivetrain can be seen in figure 13. Here we can see how the torque is increased where it is needed most. As was mentioned in earlier chapters, a heavy work machine with heavy load often needs torque at low speeds. This is something that a combustion engine typically can´t provide because of its characteristics, but again the electric motor provides its full torque at the low speeds. Figure 13 presents the torque for different speeds of the drivetrain. Here we can see that there are three different conditions of the drivetrain; low speed, mid-range and high speed. At low speed the electrical motor is dominating; in the mid-range both electrical motor and combustion engine gives the same amount of torque; and at high speed the electrical motor will not dominate but still gives additional torque to the system. This is the reason why the diesel engine can be downsized in parallel hybrid drivetrains while the drivetrain still produces at least the same amount of torque as a conventional drivetrain.

2.1.5 Power split or Series-parallel hybrid drivetrain

According to (Reif et al., 2011) a series-parallel hybrid drivetrain is made by making a connection between the generator and the electric motor, seen in figure 9, with a mechanical clutch. This way the series-parallel drivetrain can act as a series hybrid at low speeds, utilizing its advantages, and as a parallel hybrid ignoring the series hybrids disadvantages at high speeds (Reif et al., 2011). The series-parallel hybrid drivetrain is, on the other hand, more complex, as can be seen in the principal layout of the drivetrain in figure 14. This way the advantage of compactness in series hybrid drivetrain is lost. Also another electric drive is needed compared to parallel hybrid drivetrain.

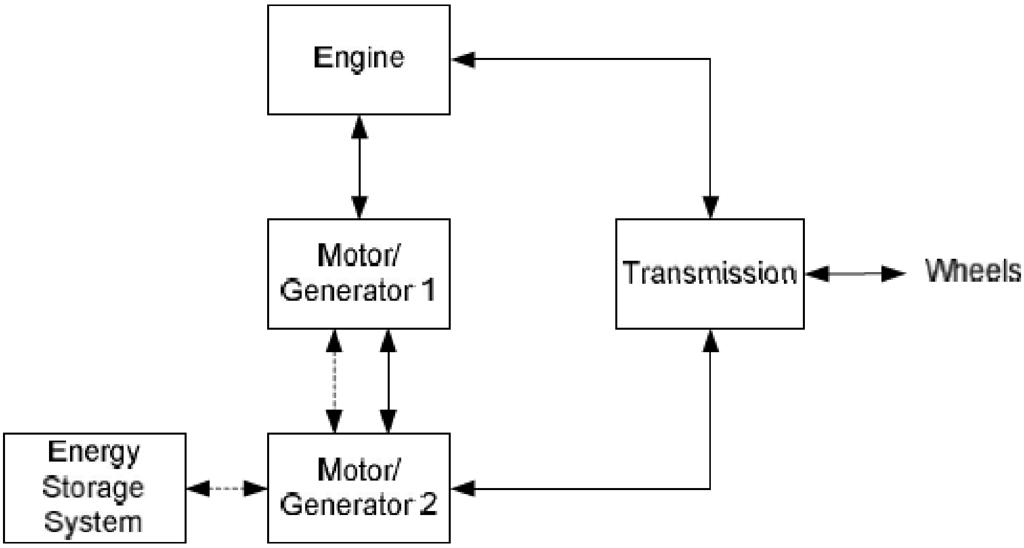


Figure 14. Generic Series-parallel combination architecture. (Zhou, 2011)

The principle of the power split hybrid drivetrain is similar to a series parallel hybrid drivetrain, combining the advantages of both series and parallel hybrid drivetrains. According to (Reif et al., 2011) a power split hybrid is always a strong hybrid, since all the required functions are possible, see table 1. The revolutionary technology in power split hybrids is in the power splitting device seen in figure 15. This device allows combining of several power sources into one, but gives also interesting possibilities to control the drivetrain with different combinations of the combustion engine and the electric motor rotational speeds. For a thorough explanation of the working principle of a power split device please see "Power Split Device Explained" -video (Hybrid Synergy Forum, 2014). A power split hybrid can either be a single-mode hybrid with one planetary gear set, or a two-mode hybrid with two planetary gear sets (Reif et al., 2011). The single-mode hybrid is more common but there are some advantages in the two-mode hybrid considering fuel consumptions and electric complexity (Reif et al., 2011).

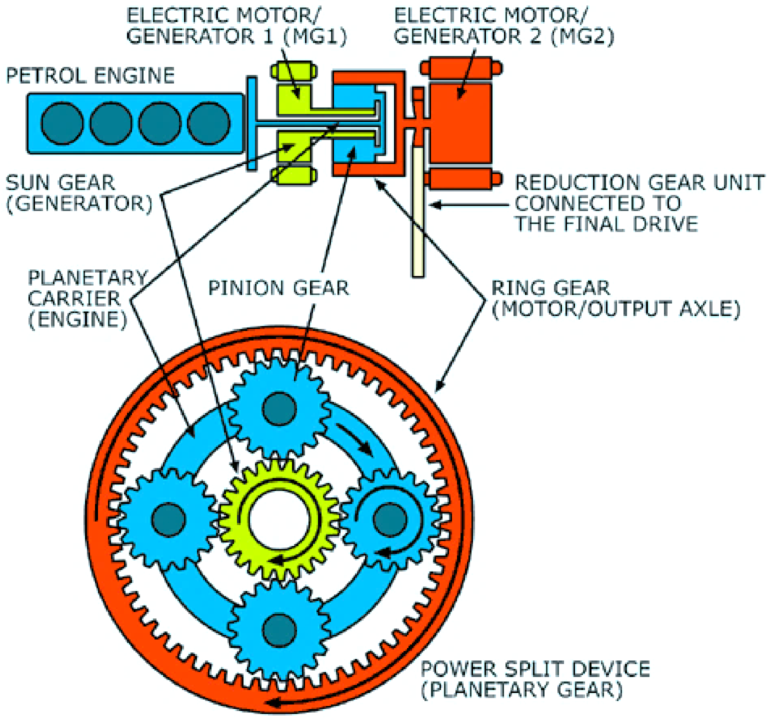


Figure 15. The power splitting device of the Series-parallel Hybrid drivetrain. (Narasimhan, n.d.)

Toyota has a well-developed hybrid drivetrain in their “Hybrid Synergy Drive” (Toyota Motor Corporation, 2014). Toyota has been one of the first on the market with hybrid drivetrain and they have gained a great number of patents on different technologies in the field of hybrid drivetrains. According to (Blackman, 2014) this number is more than 4,000. Toyota´s power split device is a high end product that is sold probably in millions. Although these kind of products exist, in the design case of this thesis a more flexible transmission is needed that might be produced in less than one thousand. Therefore, a power split device is not a solution.

3 Design perspective

In the earlier chapter hybrid drivetrains have been considered on a general level in both passenger vehicles and work machines. From this chapter and forward only non-road mobile machinery will be considered in this thesis.

The design process of hybrid or electric NRMM is similar to the design process of a NRMM with conventional drivetrain. In both cases the design process follows the v-model design path explained for example by Ziemniak et. al. (Ziemniak, Stania, & Stetter, 2009). However, there is a great difference between the designs of the different type of drivetrains on a subsystem and component level. In order to be able to implement the different subsystems and components that are needed in hybrid- or electric drivetrains, there is a great need of interdisciplinary knowledge. In hybrid drivetrains many different sciences are combined such as mechanical engineering, electric engineering, internal combustion engine technology and information technology. When the level of complicity increases, there is also a higher demand put on the user interfaces so that somebody who knows how to operate a traditional work machine could easily learn to operate the more advanced one. This chapter is an introduction to some of the main characteristics and challenges that can be encountered in the design process of the different subsystems and components in a hybrid NRMM drivetrain.

3.1 Batteries

Battery technology is at the moment the demanding field in hybrid drivetrains. There are a number of challenges encountered for the battery manufacturers such as weight to energy capacity ratio, price to energy capacity ratio and weight to power ratio. These challenges have a great effect on the design process since the cost effects the payback time for the machine and the added weight effects the amount of payload the machine can handle and thereby the productivity of the machine. There has also been discussion about how to handle defected batteries. Does the whole battery need to be replaced or is there a possibility to only change a single defected cell? This becomes more interesting towards the end of the lifetime of the machine when investing in a whole new battery is unprofitable. For conventional drivetrain components it is common to replace and maintain even minor components.

When considering weight and size of the battery pack, it is good to keep in mind the center of mass. There can be a risk of the center of mass rising too high if the battery pack design is located too high. For NRMM integration of batteries is one option, but it is expensive and results in more complicated maintaining or replacing of batteries. Modular batteries are cheaper to implement especially for small production series NRMM. Also maintaining of modular batteries is easier. It is good to consider whether it would be wise to change drivetrain architecture if the battery is getting relatively expensive or too heavy. Different types of hybrid architectures typically need different type of batteries.

Batteries operate most efficiently on a certain level of temperature. For some cases it is necessary to design a cooling or heating system for the battery pack. In some applications both cooling and heating are needed. This is good to take into account when stacking the battery cells. Considering battery heating it is a good rule to have the battery pack oversized in terms of power for the battery pack not to overheat. A battery pack always needs Battery Management System (BMS) to limit the output and input electric current of the battery. With a good designed BMS the battery will never take nor give more than it can handle. The BMS also takes care of balancing of cells i.e. evens the voltage differences between the different cells. Advanced battery management systems can also provide additional features such as battery State of Charge (SOC) or health monitoring.

3.2 Electric motor

The electric motors and generators in a hybrid drivetrain are always determined on the basis of the requirements of the performance of the NRMM. When choosing an electrical motor, it is necessary to consider the battery performance. An oversized motor will not cause any damage to the battery if there is a BMS that limits the discharge current of the battery. If there is an oversized generator there is a need for control of the generator for the charging current not to be too high for the battery. If there is a risk of the charging current exceeding the limit of the BMS, the system design needs to direct the overvoltage caused by the too high current. Solutions for this are brake resistors or capacitors.

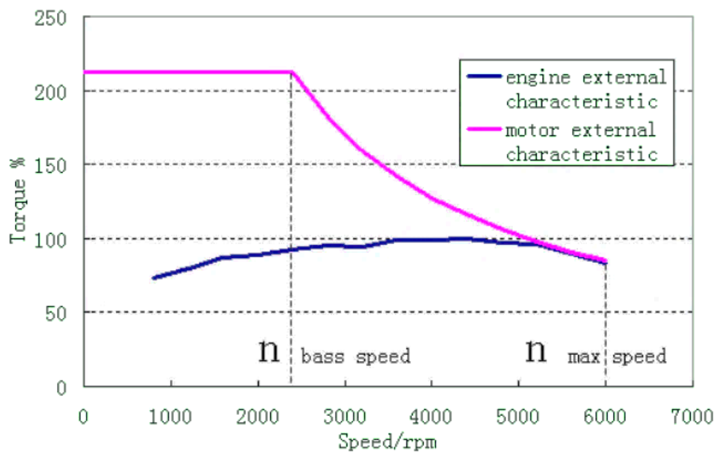


Figure 16. Comparison of external characteristics of engine and motor. (Chuan, Minghui, & Ziling, 2013)

When designing a drivetrain where an electric motor replaces the combustion engine or has to work together with it, there are differences in the characteristics between these two power sources that are worth considering. A combustion engine and an electric motor with the same maximum power do not have the same performance. This is caused by the difference in the torque behavior of these two, seen in figure 16. In the figure the torque is given as percentages of a combustion engines maximum torque. According to equation (2) (Dietsche et al., 2002) the maximum power is the same for both the engine and motor in figure 16 since the torque is the same at the maximum speed. However, the motor torque is many times greater than the torque of the engine at low speeds. An electric motor is also able to be overloaded for a short period of time. This additional power capability is good to take into account in the design process.

$$P = M * \omega = M * 2\pi * n \tag{2}$$

The purpose of an electric motor or a combustion engine in a drivetrain is to provide traction force to the wheels. This can be implemented in a hydraulic or mechanical way. The traction force is straight proportional to the torque of the power source. Traction force for most of the heavy NRMM is needed most on the low motor speeds when starting. Therefore, when designing a replacement of a combustion engine with an electric motor, it is good to consider the traction force needed and that way calculate the needed performance of the electric motor. If the electric motor will operate alongside a combustion engine in a parallel hybrid, it is good to consider in the design if these two are able to work together.

A problem that might occur when building smarter drivetrains and drivetrains with more electricity involved is electromagnetic interference. This is a good thing to consider when designing wiring and fastening of the electric components, such as sensors and motors. There is a need of grounding and isolation in order to avoid phantom faults.

The electric motor needs cooling, just as do the batteries, but in a different temperature range. The cooling element can be designed to be air or liquid. In the design process it is good to consider that the more power is taken out from the motor the more cooling it needs. Especially overloading the motor is allowed just for a certain amount of time because the motor will overheat otherwise. Better cooling results in higher power output of the motor.

The efficiency of combustion engines is rather low especially when operating on partial load. Depending on the operation point of the combustion engine, its efficiency can also vary enormously. When designing hybrid drivetrains it is good to take into account that for electric motors, the efficiency is much higher and this variation is much smaller. Therefore the electric motor can easily be operated on a wider range of operation points than a combustion engine. Controlling of electric motors is also much faster and easier. The controller can be intelligent which helps building up safety functions and raising the level of automation.

There are several different types of electric motors and they have different rotational speed ranges. Therefore, depending on the electric motor chosen for a certain drivetrain, there is a varying challenge put on the rest of the rotating parts in the drivetrain, especially before down gearing. The main challenge for rotating parts is that they need to be mounted via bearings to rigid housings. These bearings need lubrication and at high rotational speeds the angular acceleration causes the lubricating medium to escape in a radial direction and does not stick at the point where the lubrication is needed. Also high rotational speeds are a problem for bearing shielding. The higher rotational speed the less shielding is available. This is because all bearing shields cause friction at the moving seam. Therefore the higher the rotational speed the wider the gap needs to be at the seam and the bigger particles are allowed into the bearing. These are important matters to take into account when choosing an electric motor in the design process. A good electric motor has built-in bearings that take both axial and radial forces. Sometimes there is a risk that bearings will break down due to currents through bearings caused by shaft voltage. This is worthwhile to consider especially in cases of bigger electric motors.

3.3 Transmission

In a traditional drivetrain the transmission takes care of the gearing between the wheels and a combustion engine or alternatively a hydraulic motor. In a hybrid or electric drivetrain the transmission has to take care of gearing between an electric motor and the wheels, a combustion engine plus electric motor and the wheels or in some cases even a combustion engine plus several electric motors and the wheels. The level of complexity is high for transmissions in power split hybrid drivetrains. Combining several power sources can be solved for example with a planetary gear as mentioned earlier in chapter 2.1.5.

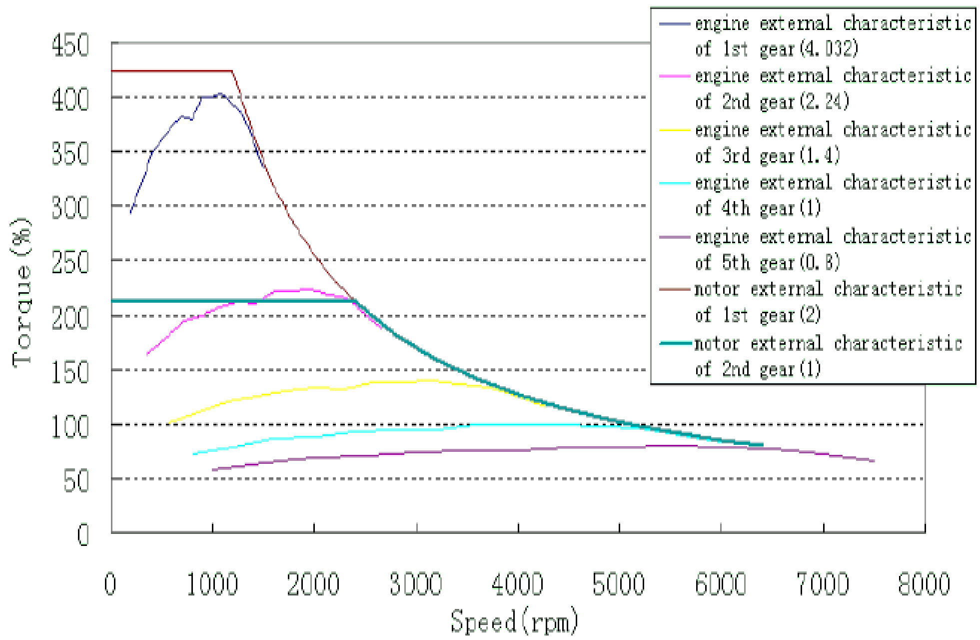


Figure 17. The comparison of external characteristics between motor with two gears and engine with five gears. (Chuan et al., 2013)

When designing transmissions for electric motors, it is good to consider the differences in torque and efficiency characteristics for combustion engine and electric motor as mentioned in the previous chapter. Because of these differences in torque and efficiency behavior, the electric motor does not need as much gearing as a combustion engine. Figure 17 shows the torque for each gear for a two geared electric motor and a five geared combustion engine. The torque is presented as percentages of the maximum torque of the combustion engines with a straight gear. As both the electric motor and combustion engine are of same maximum power, we can see that the electric motor torque curves form a nice slope joined by the combustion engine curves. This figure shows us how nicely an electric motor can cover a wide range of operation points with just few gears.

The high rotational speeds and torque of the electric motor are challenges when designing transmissions. Electric motors typically favor running at higher rotational speeds than combustion engines and still provide a great amount of torque at low speeds. Transmission parts provided by many manufacturers are able to handle the torque and speed characteristics of a combustion engine but not necessarily the ones of an electric motor. When designing hybrid and electric drivetrain it is good to consider matters such as bearing life, balancing of rotating parts, inertial forces, and lubrication and tension forces in the case of belt or chain drive. These matters are important when designing traditional drivetrains as well. However, even though the high rotational speed of the electric motor is a challenge, it is also a great advantage. When the motor rotational speed is down geared for the wheels, the torque that results as the traction force is multiplied relatively to the down gearing. This is good to take into account especially when designing drivetrains for slow NRMM. The down gearing allows choosing an electric motor with less torque. This typically means less mass and smaller face width of the motor. The smaller the motor the less copper is needed and the cheaper it is to manufacture.

3.4 ECU and Inverter

There is saying that poor or lacking hardware can be fixed with good software. There is some sense with this but the best option is to finalize a good mechanical design with reasonable software and calculation power. A combustion engines Engine Controlling Unit (ECU) is usually much lighter than the inverter that an electric motor needs. The space saved when changing a combustion engine to an electric motor can be taken up by the inverter. It might be easy to replace the combustion engine with an electric motor but to integrate the inverter in the same space can be more demanding. The battery pack takes up a lot of space as well. These factors need to be considered in the design.

The efficiency which can be achieved with the relevant hybrid drive is dependent not only on the hybrid topology but also crucially on the higher level hybrid control (Reif et al., 2011). There are different operation strategies for hybrid drivetrains. These operation strategies determine how the drive power is shared between the electric motor and the combustion engine. The strategy decides the extent to which the potentials for fuel saving or for reducing the emissions of a vehicle are utilized (Reif et al., 2011). The operation strategy determines also how regeneration is utilized. Different strategies can aim to reduce a certain emission. There is, for example, one for reducing NO_X and one for reducing CO_2.

4 Comparison of power transmissions

There are a number of different types of power transmissions. The aim of this thesis is to briefly introduce the reader to the most common types of transmission. The characteristics of the different transmissions are introduced in this chapter. Also as a concept level design guide, this chapter gives the reader some information on what kind of matters need to be considered for different types of power transmission. There are no exact rules and for some manufacturer´s parts some matters such as maintaining objects and frequency need more careful consideration.

According to (Lechner & Naunheimer, 1999) a vehicle transmission is defined in the following way: “The function of a vehicle transmission is to adapt the traction available from the drive unit to suit the vehicle, the surface, the driver and the environment”. This means that the transmission is an interface between the power source and the environment. A transmission has both technical and economical requirements and has impact on the reliability, fuel consumption, ease of use, road safety and transportation performance of the vehicle. (Lechner & Naunheimer, 1999)

A few of the most common power transmission types are introduced and evaluated. These power transmission types are gearwheel drive, chain drive and belt drive. The characteristics of each power transmission technology will be introduced in their own chapters. The aim of this chapter is to consider different options when there is a need of a single speed gearbox for fitting the electric motor to the rest of the drivetrain in an electric or hybrid drivetrain.

4.1 Gearwheel drive

In gearwheel drives the transmission flow is normally between parallel shafts, using spur-toothed and helical-cut spur gears (Lechner & Naunheimer, 1999). According to (Lechner & Naunheimer, 1999) by far the greatest proportion of vehicle transmissions are gearwheel transmissions. Gearwheel drive has the highest power to weight ratio for converting speed to torque (Lechner & Naunheimer, 1999). Incoming versus outgoing shaft can also be in different angles and directions. Different types of gearwheel drives are presented in figure 18 as following:

a) Cylindrical gearwheel drive with straight teeth
b) Cylindrical gearwheel drive with skew teeth
c) Cylindrical internal teeth planetary gearwheel drive
d) Gearwheel and gear rack
e) Bevel gearwheel drive
f) Skew spur wheel drive
g) Worm and worm gear

Gearwheel drives need plenty of lubrication. There are different levels of lubrication such as splash type lubrication, bath lubrication and flood lubrication. Because of the lubrication, gearwheel drives always need be covered by a waterproof cage. Gearwheel drives do not usually need any other kind of maintenance during their lifetime other than lubrication oil checking and perhaps changing every year or every few years. At high rotational speeds the lubrication might not work the same as for low speeds due to the angular acceleration that pushes the lubrication medium out from the center of the gear wheels. If the entire lubricating medium is forced towards the sides of the cage, there is a great risk of breakdown.

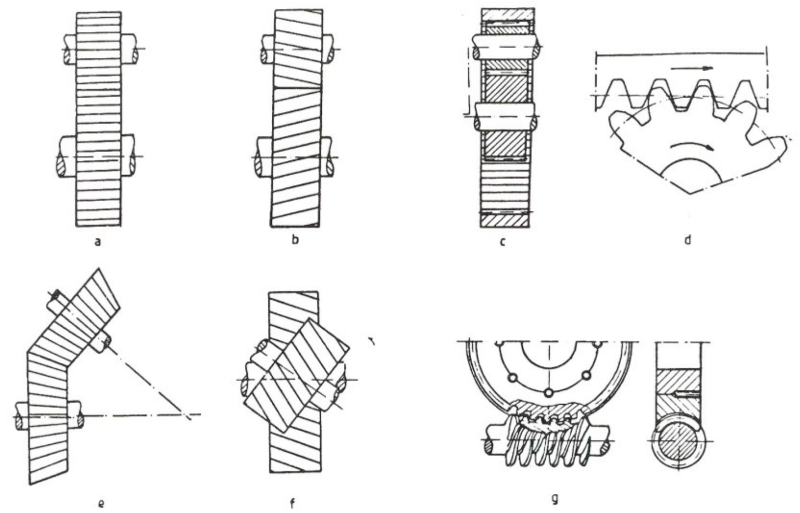


Figure 18. Different types of gearwheel drives. (Airila et al., 1985)

Assembling of gearwheel drives is simpler than for chain or belt drives. Gearwheels need to be put in the right order on right places and assembly is ready. Since the gearwheels stick next to each other there is no need of finding a proper tension for the connection. However, the manufacturing of the casing for the gearwheel drive is demanding because of the strict tolerances. In the design case in this thesis, a belt drive is designed that can be straight bolted on to the rest of the drivetrain. In this case assembling is made simply.

4.2 Chain drive

For chain drives the transmission flow is always between parallel axles. The chain drive works most efficiently when the axle ends are facing same direction. Reasons for this are easier mounting, operation, axial alignment, and chain tightening. For chain drives the chain gears are determined by the chain structure. Figure 19 presents different types of chains. Chains can be either of pulling or pushing type. Some Continuously Variable Transmissions (CVT) are made using push belt technology. In figure 19 the dimension “p” stands for pitch. The longer the pitch is the thicker the chain can be and the more torque can be transmitted. Increased pitch length increases the size of the chain gears in the same proportion.

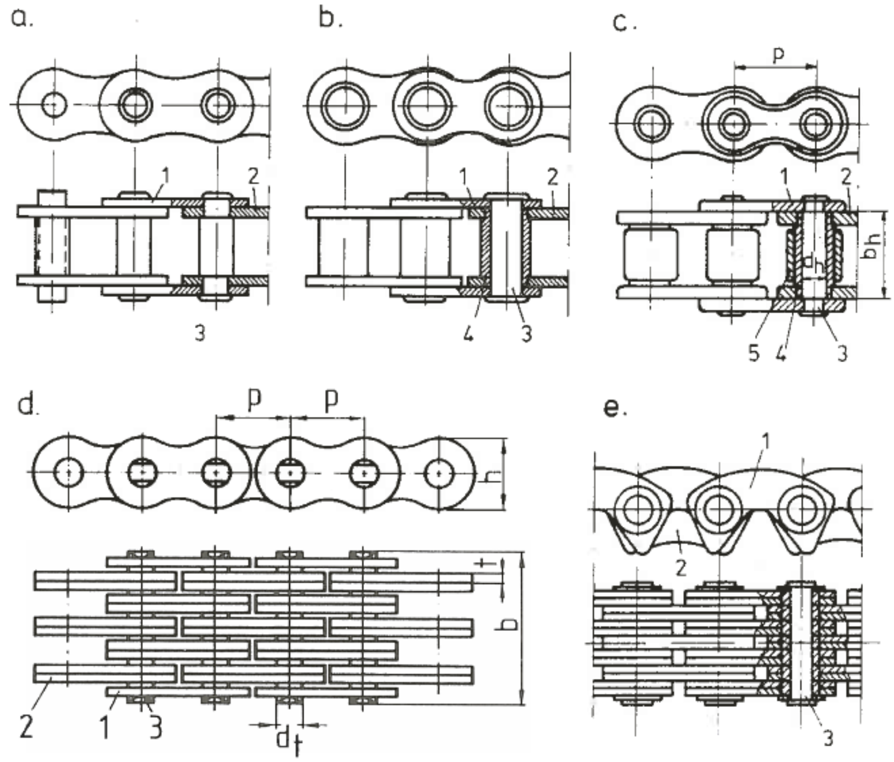


Figure 19. Different types of chains. (Airila et al., 1985)

Chain drives need less lubrication than gearwheel drives. For chain drives, the most common type of lubrication is a thin oil film over the whole chain. Also splash type lubrication is possible. The angular acceleration causes problems for chain drive lubrication as well as for gearwheel drives at high rotational speeds. This lubrication in chain drives needs maintaining, as do gearwheel drives. However, the maintenance frequency is lower if the chain drive is covered. In case the chain drive is open to all dirt, dust and water, the maintaining frequency is much higher than for gearwheel drives. To notice at this point is that chain drives do not need to be covered to be able to work. Other maintenance needed for chain drives is chain tightening. When the chain joints wear, the chain gets loose over time. Therefore chain drives need a check of proper tension every few years. Finding the proper tension for the chain causes the assembling of a chain drive to be more complicated than for gearwheel drives. Proper tension for chain drives is usually monitored by measuring the mobility of the chain on the loose side. This mobility is typically several millimeters or centimeters.

4.3 Belt drive

Belt drives have not been used for a long time in most of the higher power NRMM applications. Some motorcycle manufacturers are using belt drives instead of chain. Many years ago transmission manufacturers decided not to continue using belt drives preferring gear wheel or chain drives. Now, however, the belt drive technology has improved enormously and some manufacturers have recognized that the developed belt drives could again be a reasonable choice instead of gear wheel or chain drive. It is worthwhile to consider that belt drives have been used over this time in many smaller power applications, such as powering auxiliary devices needing power less than 20kW. This is probably the field where belt drives have always been used the most. However, there are still a great number of people that believe that belt drive is not the proper choice for a drivetrain in high power applications. One of the motivations in driving the design case in this thesis forward was to challenge this statement.

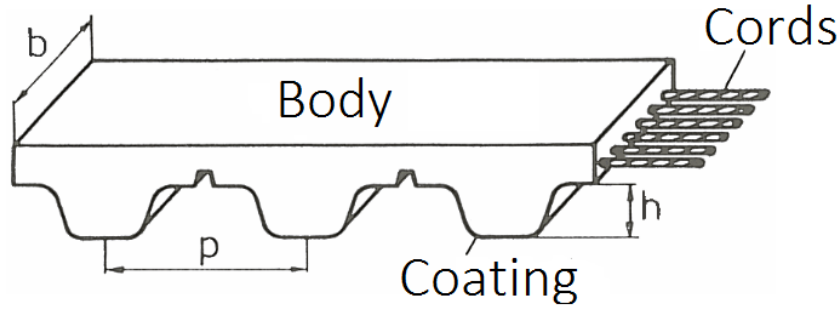


Figure 20. The structure of a synchronous belt. (Airila et al., 1985)

A belt drive consists of belt pulleys and belt. The pulley structure is determined by the belt shape. There are several different types of belts but the most common ones are v-belts and synchronous belts. Transmitting high torque with v-belt requires multiple belts in a row. This is seen many times in industry with heavy rotating machines for example in paper machine roll grinders. Synchronous belts can transmit much more torque than v-belts due to the different structure that does not allow slipping of the belt. Synchronous belt drives are lighter and take up less space and are therefore more popular especially in NRMM. Figure 20 presents the structure of a synchronous belt. There is ongoing development especially in materials used in belts. The cords seen in figure 20 for some manufacturers are made from carbon fiber (Gates, 2014), which prevents belt fatigue at high rotational speeds. Improvements have been also achieved in materials used in the belt body and coating. Currently, many belt manufacturers produce the body of polyurethane or neoprene and the coating using nylon. Previously many belts did not even have coating and the body was rubber (Lehto, 1987). This caused belt failure especially in cold conditions.

For belt drives the transmission flow is in most cases between parallel axles but there are a number of different implementations. Figure 21 presents different belt drive layouts. When the axle ends face in the same direction, it is easiest to mount the belt. The tolerances for belt tightening and axial alignment are loose for small power belt drives but when power and speed is higher the tolerances are stricter. In case of axle misalignments, there is a force on the belt in axial direction. Belt falling off is prevented in these cases by flanges on both sides of the pulley but the belt will get worn out from the sides if this happens too much. As opposed to gearwheel drive and chain drive, belt drives do not need any lubrication. This allows the belt to operate unmaintained for a longer period of time. This means also that the same lubrication issue presented for gearwheel and chain drives at high rotational speeds with the angular acceleration pushing lubricating medium away is not affecting belt drives. This is a great advantage for belt drives when considering high rotational speeds.

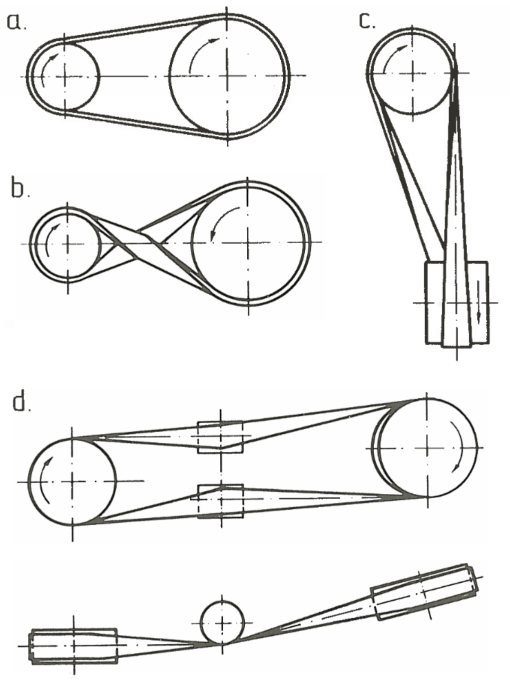


Figure 21. Different types belt drive layouts. (Airila et al., 1985)

Belt drives do not necessitate a cage as cover, but protecting the belt drive from dust, dirt and water increases the lifetime of the belt. Belt drives need regularly checking in terms of belt tension for the belt not to be able to slip if it becomes loose.

Assembling of belt drive is demanding for the high power range drives because of the high tension forces. However, this can be performed in field conditions with proper tools. A belt drive needs also tensioning but instead of monitoring the tension by measuring a distance the belt tension is usually measured with acoustical measurement devices or with force sensors. A good belt tensioning measurement device can be expensive for the consumer to buy. This is an additional cost compared to the other types of drive. However, with a good acoustic tension measuring device, failure diagnostics of the belt can also be performed. This way belt breakdown can be prevented by replacing the belt before it breaks down.

A belt does not stretch significantly but acts in an elastic way. This is an advantage when considering coupling for the drive since the elasticity in the belt drive compensates the need of elastic coupling in many cases. Gearwheel and chain drives always need elastic coupling to soften the peak loads. In the case of an electrical motor, an elastic coupling is not necessarily needed.

5 Tubridi design case

In this thesis work, a design case was performed. This design process as a whole was the most rewarding part of the entire thesis work. In the design case engineering basic knowledge in many areas were utilized on a practical level.

The design case was performed as a part of the Tubridi-project. The case Load-Haul-Dump (LHD) machine in the Tubridi-project is a 14 ton underground mining loader of model EJC 90. The purpose of the Tubridi-project is to demonstrate new technologies and technical solutions for future NRMM accomplished mainly by implementing hybrid drivetrain technology. Figure 22 presents a typical traditional drivetrain for an underground mining loader. In the figure we can see that the internal combustion engine is providing both front and rear axles with mechanical force through a drivetrain consisting of mechanical components. Transmission from combustion engine to differentials can also be hydraulic. The demo machine is an underground mining loader. It is basically a NRMM with articulated steering. This is a relatively general layout that can be found in some tractors and loaders.

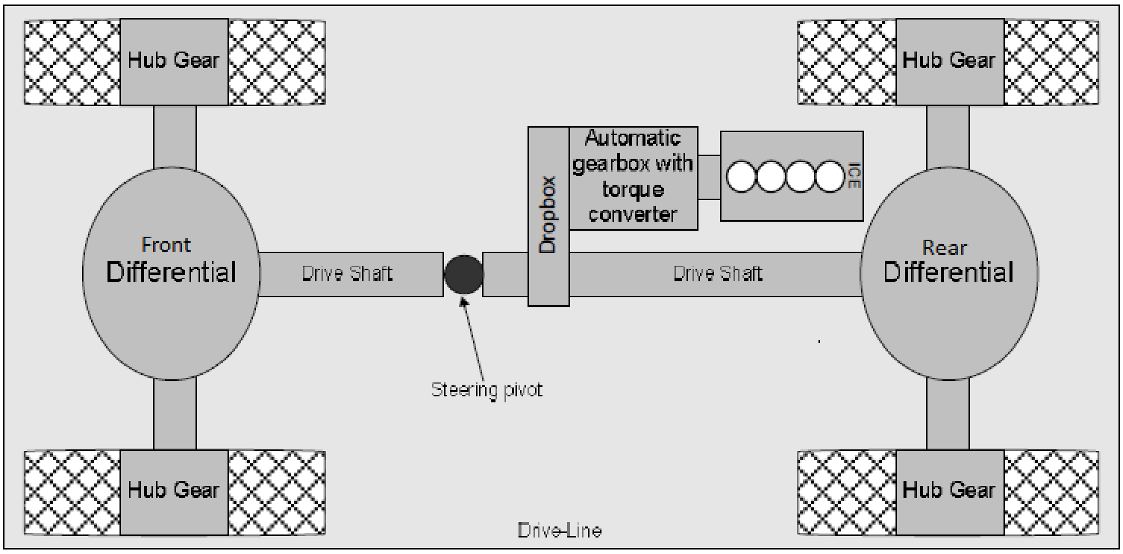


Figure 22. Typical conventional drivetrain for mining loader. (Lehmuspelto, Heiska, & Leivo, 2010)

The duty cycle of the underground mining loader considered in this thesis is presented in figure 23. In this duty cycle we can see that there is a great potential for energy regeneration caused by the elevation of the driven track. Figure 24 shows the principal idea of the hybrid mining loader duty cycle.

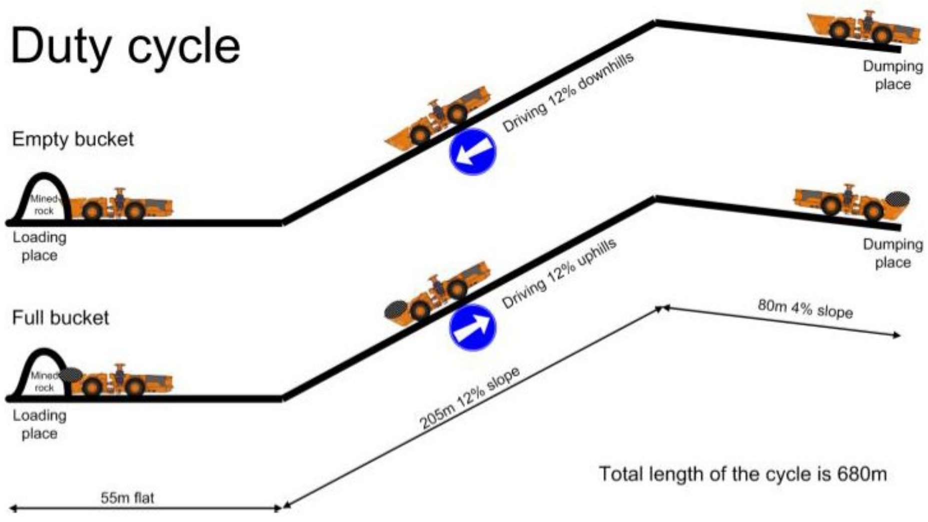


Figure 23. The duty cycle of the underground mining loader. (Lehmuspelto et al., 2010)

There is a slight difference in the duty cycles presented in figures 23 and 24. There is yet an additional potential of regeneration seen on the rightmost slope in figure 23 that is not included in figure 24. However, the idea is the same.

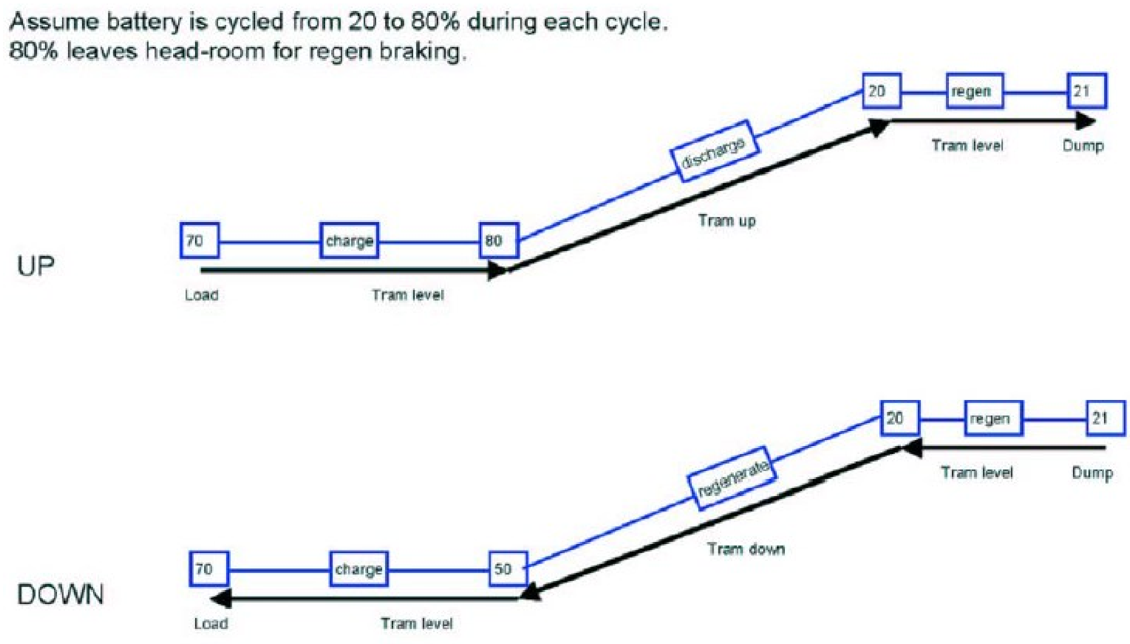


Figure 24. The duty cycle of a hybrid mining loader. (Barnes, 2004)

Previous research (Barnes, 2004) on hybrid underground mining loaders, figure 25, shows us that converting a conventional underground mining loader to a hybrid one is reasonable. This R1300 duty cycle presents the power and energy measurements for a fuel cell hybrid underground mining loader but works as well for a diesel hybrid drivetrain. The duty cycle is divided into a number of jobs and the figure shows the mean power of each job and total mean power as well as actual power and the cumulative energy over the whole cycle. As we can see in the figure, with a conventional drivetrain the combustion engine would need to be designed by the peak power need i.e. 200 kW. In the case of a series hybrid drivetrain, the combustion engine can be designed just above the mean kilowatts i.e. 50 kW and the battery can provide the peak power.

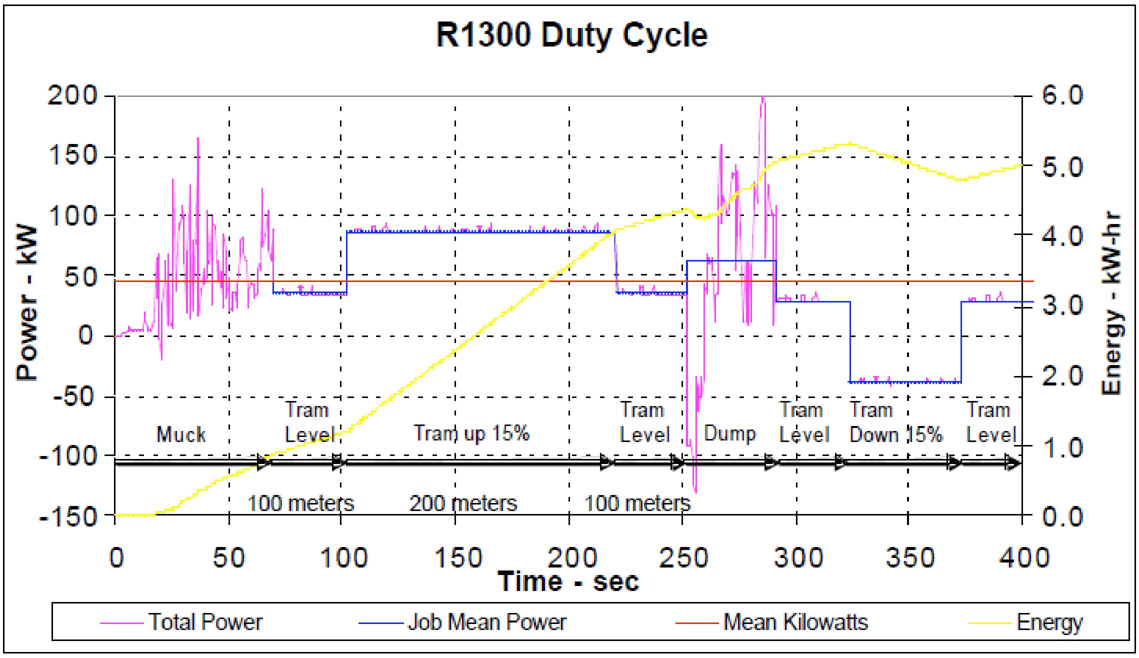


Figure 25. R1300 Duty Cycle of the mining loader. (Barnes, 2004)

In the Tubridi-project the mining loader is equipped with an electromechanical drivetrain. This means that the power flow from the energy source travels to the point of drivetrain where traction force is needed in two different energy forms. First there is a downsized internal combustion engine attached to a generator that transfers the mechanical energy from the internal combustion engine to electric energy. This electric energy is stored in batteries. From the batteries the electric energy flows to the two electric motors, one on each axle, to generate the needed speed and torque for the rest of the drivetrain. Behind each electric motor, one belt transmission is mounted to bring the rotational speed of the electric motor down to the sufficient speed for the rest of the drivetrain. This way the needed torque is also achieved when down gearing. On the rear drivetrain for the rear axle there is a three-speed gearbox in addition to the belt drive. The Tubridi-project drivetrain configuration can be seen in figure 26. In this drivetrain, only components from electric motor inverters to wheels are included. The rest of the drivetrain components, such as internal combustion engine and generator are not considered in this design case.

In the design case included in this thesis the main focus is on the belt drives seen on both front and rear axles in figure 26. This also includes the interfaces on both sides of the belt drives. Belt drive technology was chosen for this application due to its great advance during the past years. Another reason for choosing this technology as a transmission solution was the cost efficiency of belt drives. This is especially significant for machine manufacturers that do tailored engineering for a low number of power transmissions per year. Engineering a power transmission for a tailored case with high power and speed might require a great number of purpose built components. This case study is a trial to find out how many ready-made components are available. Considering the maintenance, belt drive technology has a profit when it does not need any lubrication. Capability to run at high rotational speeds is also a certain advantage.

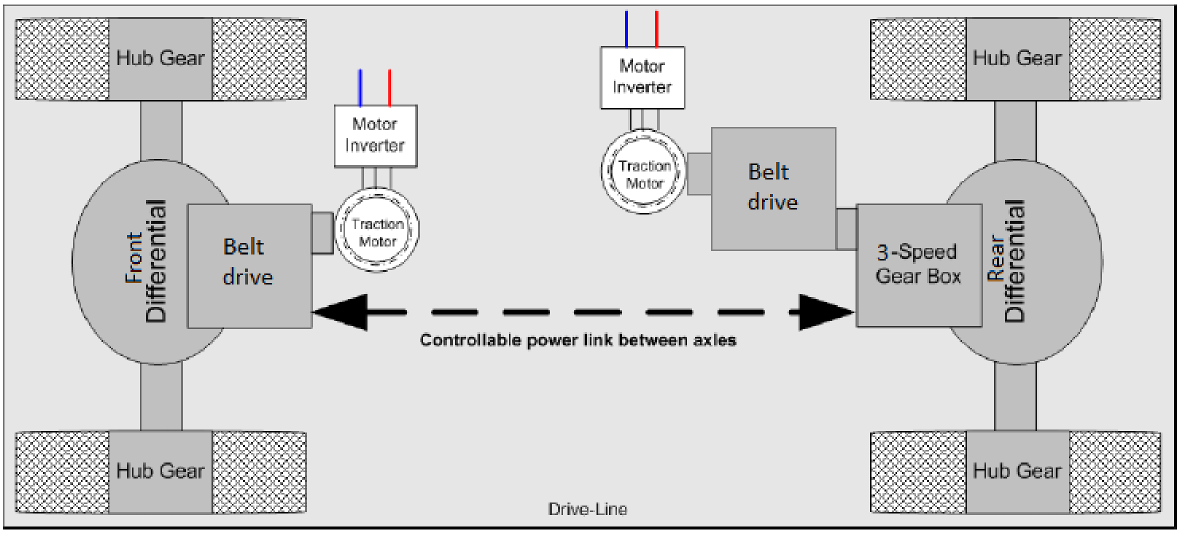


Figure 26. The drivetrain configuration for the hybridized mining loader including components from electric motor inverters to wheels.

Other solutions, such as in-wheel motors and next to wheel motors, were also considered for this project at the initial phase. However, at an early stage, the drivetrain configuration shown in figure 26 was chosen due to its cost efficiency compared to the other solutions. The other solutions would also have needed remodeling of axles and wheels, which would need more complicated designing. In the present differentials, there is a gear ratio of 5,125 and in the hub gears there is a gear ratio of 6. These gear ratios are really helpful when there is a great amount of traction force needed at the wheel. With these gear ratios the torque gets multiplied. If we would consider in-wheel motor or next to wheel motor, we could not have the benefit of these gear ratios and we would need a much bigger electric motor to be able to generate enough torque.

5.1 Initial situation

Figure 27 shows the mining loader in its initial state when the design case was begun. The mining loader looked sparse at the time since many other people were working on different parts of the loader in the Tubridi-project.

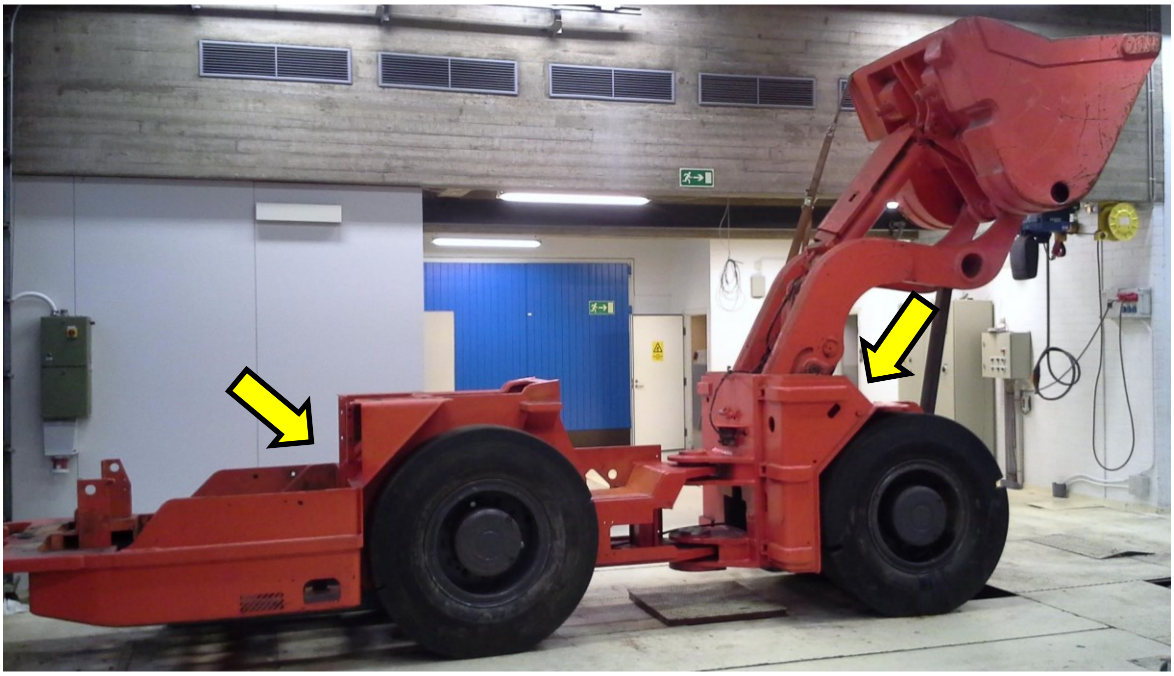


Figure 27. Mining loader at the beginning of the thesis work.

In figure 28 we can see the space reserved for the drivetrain for the front axle. This is considered a very small volume. The picture is taken from the direction shown by the yellow arrow on the right side of figure 27. In the figure 28 we can see the differential input shaft down in the middle. In the beginning of the design case, an electric motor was already chosen and delivered for the front axle drivetrain. Also the other parts, excluding the belt drive, were designed and ordered in advance.

Figure 28. Space reserved for the front axle drivetrain.

The front axle and rear axles are slightly different. The front axle is rigid and the rear axle is oscillating due to the axial connection seen in figure 29 upper part. The space planned for the rear drivetrain can be seen in figure 29. The picture is taken from the direction shown by the yellow arrow seen in figure 27 on the left side. The differential input is seen in the middle of figure 29. This differential input was pointing forward in the original drivetrain so the axle needed to be removed in order to turn it around. This is because the electro mechanic drivetrain was designed to be on the rear side of the rear axle.

Figure 29. The space for the rear drivetrain.

5.2 Design process

Required parameters for the belt drive were measured and calculated in the design process. With the known parameters the belt drives were calculated using belt drive design software received from manufacturer websites. With the known parameters for the belt drives, the belt drive casing and interfaces as well as bearings were designed.

5.2.1 Calculations

The design process was started by calculating the traction forces and the resistance forces. The calculations in this design process were performed in order to receive good estimations for the gear ratios needed for the belt drives. A main requirement was to achieve a gear ratio for the belt drive that gets the tractive effort of the hybrid mining loader to be greater than the tractive effort of the conventional mining loader. The tractive effort curve of the considered mining loader with conventional drivetrain is presented in figure 30. With the calculated traction forces and resistance forces for the hybrid conversion, a traction force diagram was plotted. By comparing this traction force diagram with the one that was provided by the manufacturer, the needed parameters for the electromechanical drivetrains were determined.

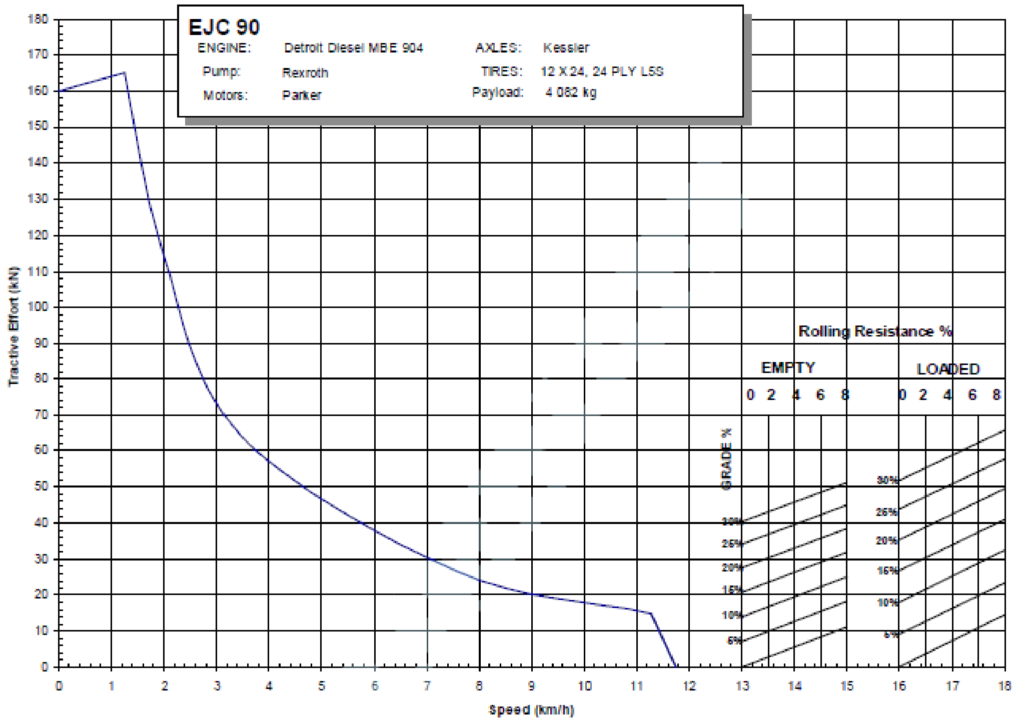


Figure 30. EJC 90 Tractive effort curve with original drivetrain. (SANDVIK, 2006)

The calculations were made with measured values and values received from the manufacturer. Not all of the values could be obtained by measuring or asking the manufacturer. Therefore, some values such as drag and rolling resistance coefficients are typical values taken from literature. First the resistance forces were calculated. The total driving resistance forces were calculated as shown in equation (3). In the equation F_r stands for the rolling resistance force, F_i for the drag force and F_n for the hill climbing resistance force.

$$F_R = F_r + F_i + F_n \tag{3}$$

The rolling resistance was calculated as shown in equation (4) (Tuononen & Koisaari, 2010). In the equation f_r stands for the rolling resistance coefficient and F_Z for the wheel load. In this design case, as the wheel load was selected to be the mass of the mining loader with the original drivetrain. The rolling resistance coefficient was estimated to be 0.03.

$$F_r = f_r \cdot F_Z \tag{4}$$

The drag force was calculated as shown in equation (5) (Tuononen & Koisaari, 2010). In the equation, ρ stands for air density, A stands for the cross-sectional area, C_D stands for the drag coefficient and v_x for the vehicle speed. The values for both cross-sectional area and drag coefficient were in this design case estimated value. As we can see in the results, the drag forces are so insignificant in this design case that they can be ignored.

$$F_{\mathrm{i}} = \frac{1}{2} \cdot \rho \cdot A \cdot C_{\mathrm{D}} \cdot v_{\mathrm{x}}^{2} \tag{5}$$

The traction force for the electromechanical drivetrain was calculated as shown in equation (6) (Tuononen & Koisaari, 2010). In the equation M_{m} stands for the torque of the electric motor, i_{tot} stands for the total gear ratio of the drivetrain, η stands for the drivetrain total efficiency and r_{k} stands for the radius of the tire as loaded. In this design case, the total efficiency of the drivetrain was a good estimation based on the efficiency of the single components in the drivetrain.

$$F_{\mathrm{x}} = \frac{M_{\mathrm{m}} \cdot i_{\mathrm{tot}} \cdot \eta}{r_{\mathrm{k}}} \tag{6}$$

The total gear ratio of the drivetrain on the front axle was calculated as shown in equation (7). In the equation i_{bd} stands for the belt drive gear ratio, i_{d} stands for the differential gear ratio and i_{hg} stands for the gear ratio of the hub gear. The gear ratio of the belt drive was the unknown term in the calculations and it was sought by iterating. The iteration was done by comparing the traction force diagrams of the electromechanical drivetrain with the original drivetrain as seen further in the results.

$$i_{\mathrm{tot}} = i_{\mathrm{bd}} \cdot i_{\mathrm{d}} \cdot i_{\mathrm{hg}} \tag{7}$$

The gear ratio of the drivetrain on the rear axle was calculated as shown in equation (8). In the equation i_{bd} stands for the belt drive gear ratio, i_{t} stands for the transmission gear ratio, i_{d} stands for the differential gear ratio and i_{hg} stands for the gear ratio of the hub gear. The differential gear ratio, as well as the gear ratio for the hub gear, is the same for both front and rear axle. The transmission gear ratio stands for the three speed transmission that is only included in the rear axle drivetrain. Therefore the transmission gear ratio for the rear axle drivetrain has three different values depending on which gear is selected. The smallest gears give additional traction force for the rear axle in the loading phase when the front axle characteristically rises up from the ground and does not provide traction force. The front axle does not need as much traction force in cases where it operates alone. Therefore a gearbox was implemented only in the rear axle drivetrain.

$$i_{\mathrm{tot}} = i_{\mathrm{bd}} \cdot i_{\mathrm{t}} \cdot i_{\mathrm{d}} \cdot i_{\mathrm{hg}} \tag{8}$$

In the beginning, an assumption was made that the rear axle belt drive gear ratio could be about 2:1 and the front axle belt drive gear ratio about 3:1. The assumptions were made on basis of the known differences of rotational speeds of the electric motor and the combustion engine. The gear ratio for the whole drivetrain was calculated by iteration. Certain boundaries where determined for the gear ratio and the best result was estimated on basis of traction force and driving speed. The boundaries were mainly effected by the physical limitations of space in the place where the belt drives were designed to be mounted. The belt drive gear ratio was able to be straight forwardly calculated from the total gear ratio calculated by iteration. The iteration was able to be performed as the other components in the total gear ratio were known. After calculations, the best gear ratios turned out to be the same as the first rough estimations. The gear ratio for the front axle belt drive was 3:1 and the gear ratio for the rear axle belt drive was 2:1. Both gear ratios are within reasonable gear ratios for a single speed belt drive. Typically a reasonable gear ratio for a belt drive is between 1:1 and 3:1. If the gear ratio would be higher, the smaller pulley would be too small compared to the larger pulley for the belt to have enough of its teeth in contact at the smaller pulley.

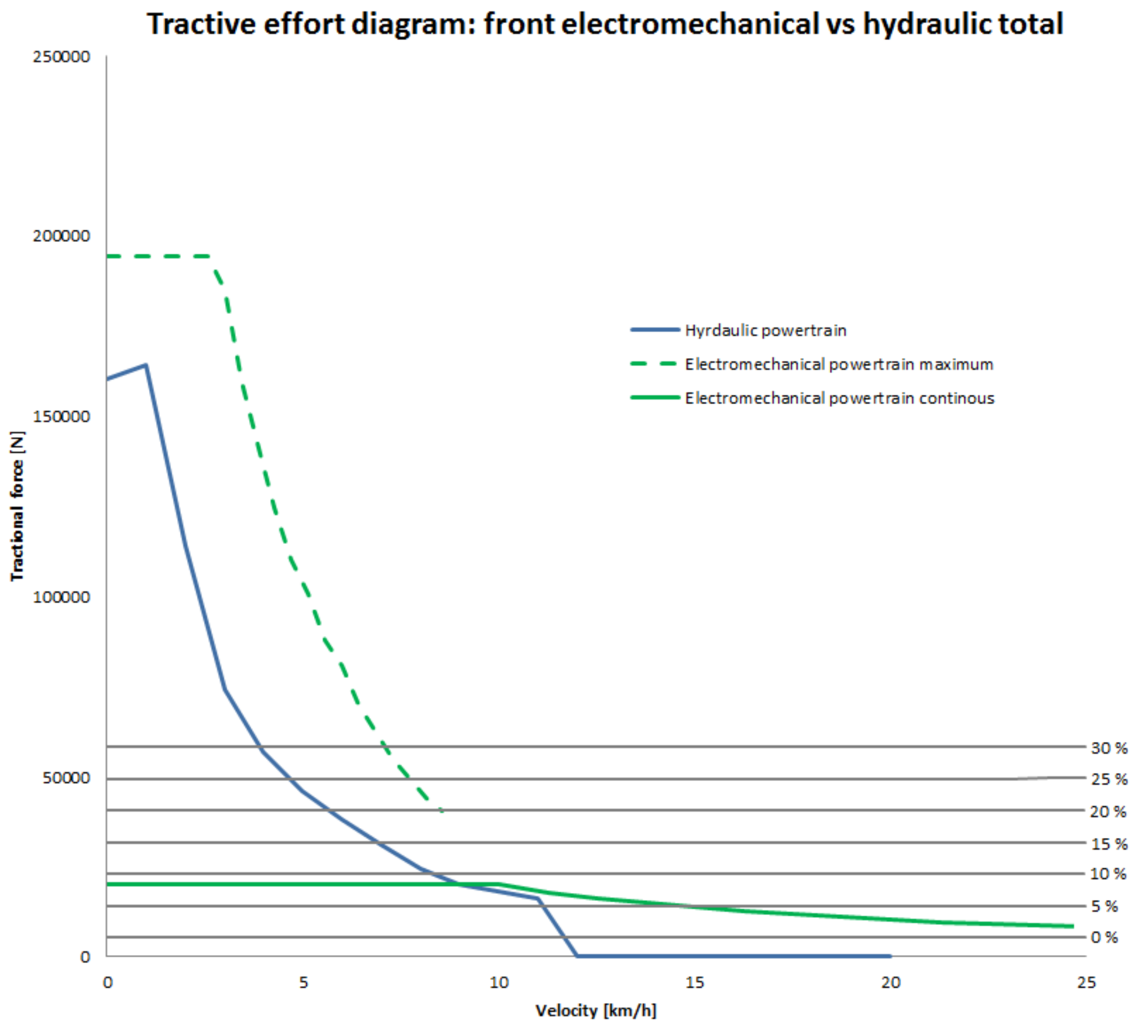


Figure 31. Tractive effort diagram for front axle electromechanical drivetrain and original hydraulic drivetrain for both axles. Grey curves indicate total running resistances with percentages of different hill steepness given in right end of each curve.

Three different tractive diagrams were created from the results calculated for the new drivetrain setup. The first tractive effort diagram is presented in figure 31. This tractive effort diagram demonstrates the tractive effort of the front axle electromechanical drivetrain when operating alone compared to the tractive effort of the original hydraulic drivetrain for both axles. The front axle drivetrain has one gear and the velocity is controlled with the electric motor frequency converter. The dashed green line in figure 31 is the maximum tractive effort that the front axle drivetrain can provide when the electric motor is overloaded. The motors that were chosen for this application can be overloaded for one minute during one hour in case of maximum overloading. This means that in this type of electromechanical drivetrain case the most demanding tasks do not have a long duration and can be covered by the increased power from the electric motor when it is overloaded.

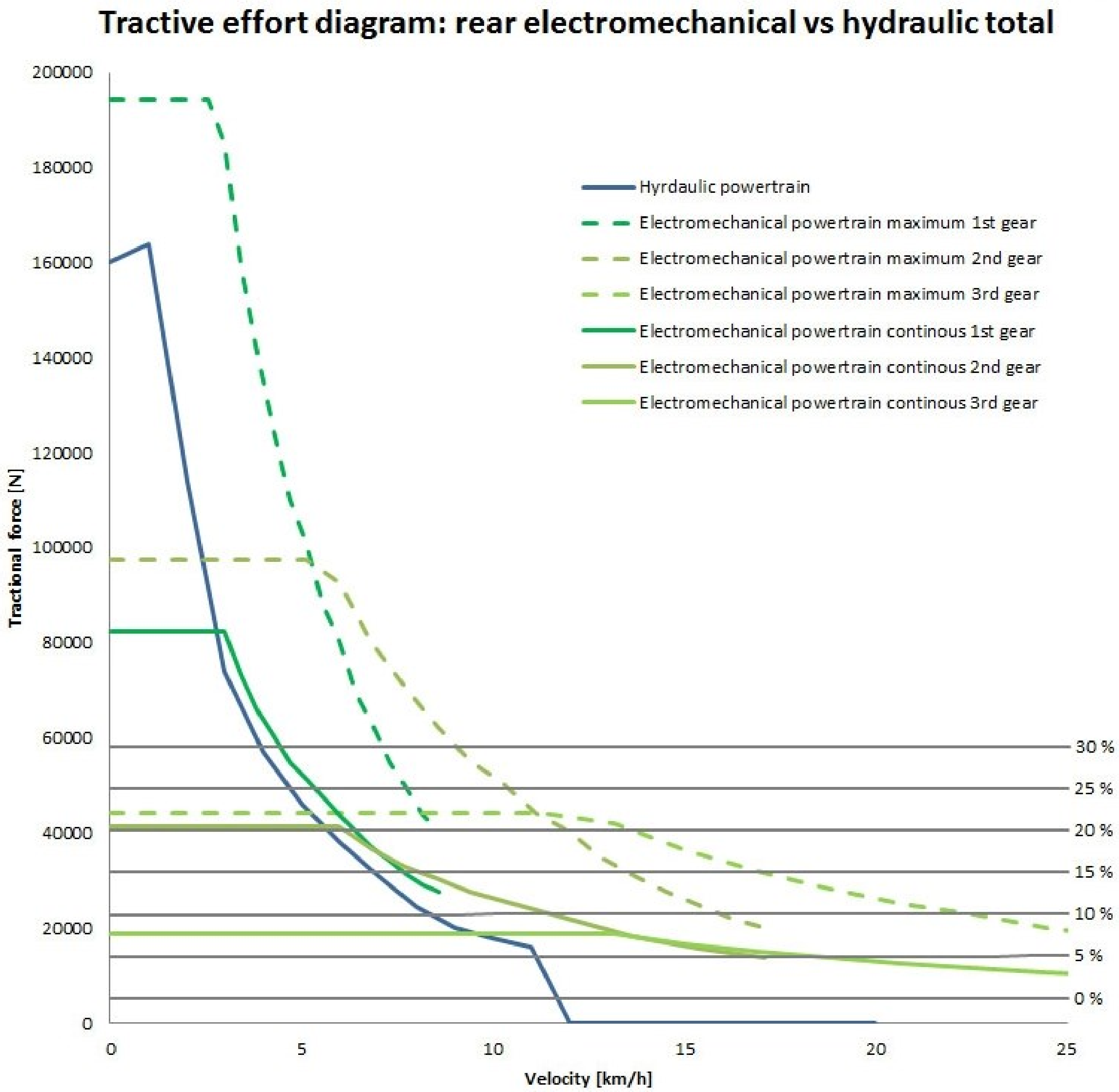


Figure 32. Tractive effort diagram for rear axle electromechanical drivetrain and original hydraulic drivetrain for both axles. Grey curves indicate total running resistances with percentages of different hill steepness given in right end of each curve.

The second tractive effort diagram is presented in figure 32. This tractive effort diagram demonstrates the tractive effort for the rear axle electromechanical drivetrain when operating alone compared to the tractive effort of the original hydraulic drivetrain for both axles. The rear axle drivetrain has three gears and using the low gears the tractive effort for the low velocities gets higher. As we can see in figure 32, the rear axle drivetrain can provide the required tractive effort alone when overloaded. As we also can see in both figure 31 and 32, for both front and rear axle electromechanical drivetrains, the velocity ranges for the electric motors are much wider than for the hydraulic drivetrain.

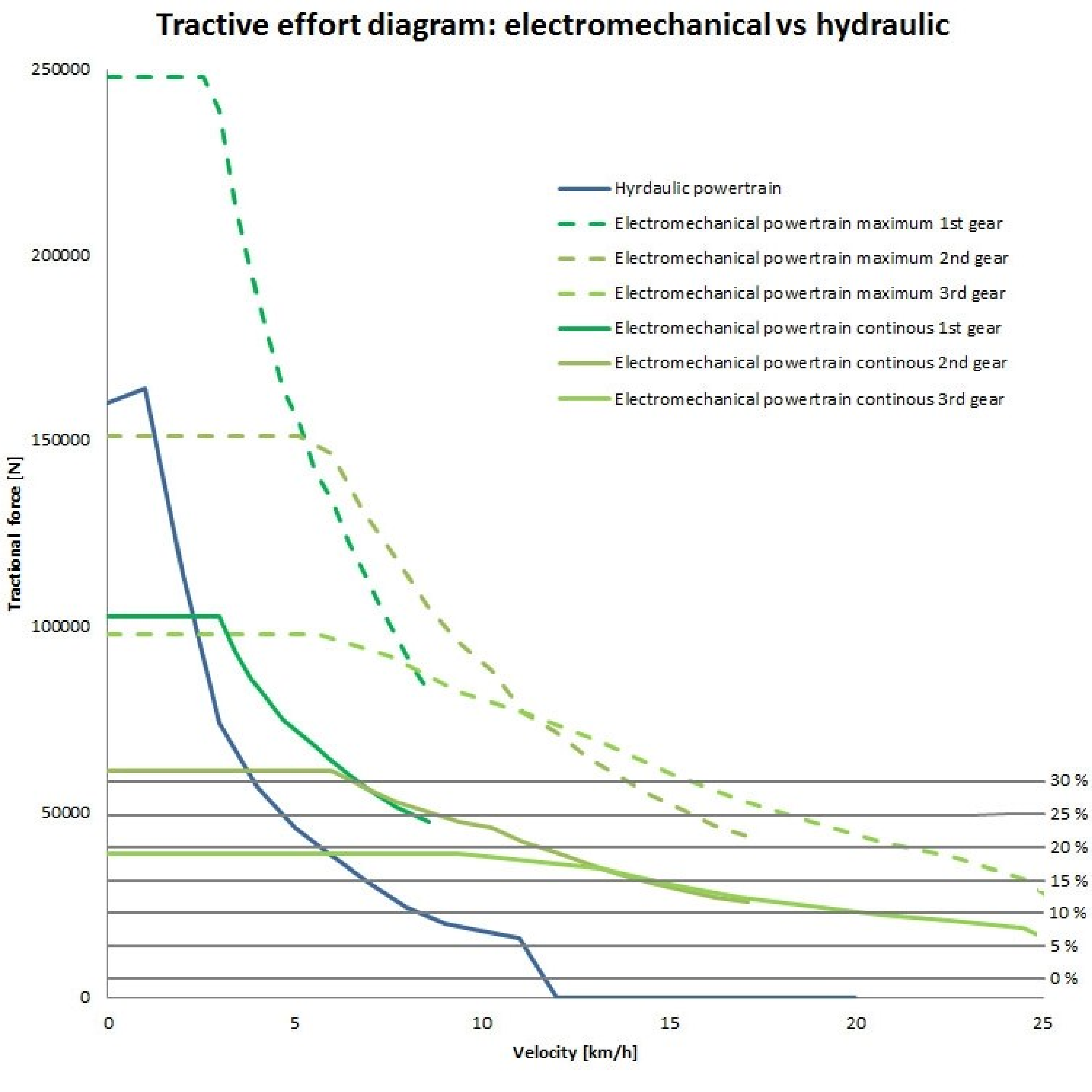


Figure 33. Tractive effort diagram for combined rear and front axle electromechanical drivetrain and original hydraulic drivetrain for both axles. Grey curves indicate total running resistances with percentages of different hill steepness given in right end of each curve.

The third tractive effort diagram is presented in figure 33. This tractive effort diagram demonstrates the combined tractive effort of both front and rear axle electromechanical drivetrain compared to the tractive effort of the hydraulic drivetrain for both axles. In the figure we can see that the new hybrid drivetrain conversion gives much better continuous tractive effort than the old hydraulic one. The only disadvantage with the electromechanical drivetrain, according to the tractive effort diagram, is that it does not give as much continuous tractive effort at the lowest velocities. However, this can be covered by overloading the electric machines.

After determining the needed speeds and gear ratios for the belt drives and taking the needed measurements for the belt drive, the design process continued by searching for belt drive manufacturers over the internet. Manufacturers such as Goodyear, Gates, Lönne, Fenner and Megadyne were considered and evaluated. The goal was to design belt drives from the advanced technologies in order to be able to evaluate the progress of the belt drive technologies. Belt drive manufacturers such as Goodyear and Gates are as the biggest players in the field. The design was considered to perform with different manufacturer belt drive parts in different ends of the machine in order to gain experience of the differences between different belt drives and manufacturers. The belt drive for the front axle drivetrain was decided to be executed in cooperation with Gates and the one for the rear axle drivetrain in cooperation with Goodyear. Gates was chosen for the belt drive in the front since they had to provide more suitable sizes of belt pulleys for this belt drive, which was crucial due to the very limited space in the front. The local retail dealers of both Goodyear and Gates turned out to be surprisingly helpful in the design process. Both of the retail dealers also provided machining of their belt pulleys to the customer needs.

5.2.2 Belt drive design by software

Some of the most common belt drive manufacturers provide belt drive design software products that can be downloaded through their internet pages. To be able to download the software one needs to register for the company internet pages. This is however free. The belt drive design software products introduced in this chapter were needed when carrying out the design case of this thesis presented in chapter 5.

Gates

With the belt drive design software, different belt drive solutions can be found for a certain purpose by giving the software a wide range of parameters. Figure 34 presents a screenshot of Gates belt drive design software parameter input window. This free software was downloaded from Gates internet page (Gates Powering Progress TM, 2014). In this software the user can select the belt models desired as seen in upper left corner in the figure. Also many other parameters are user selectable.

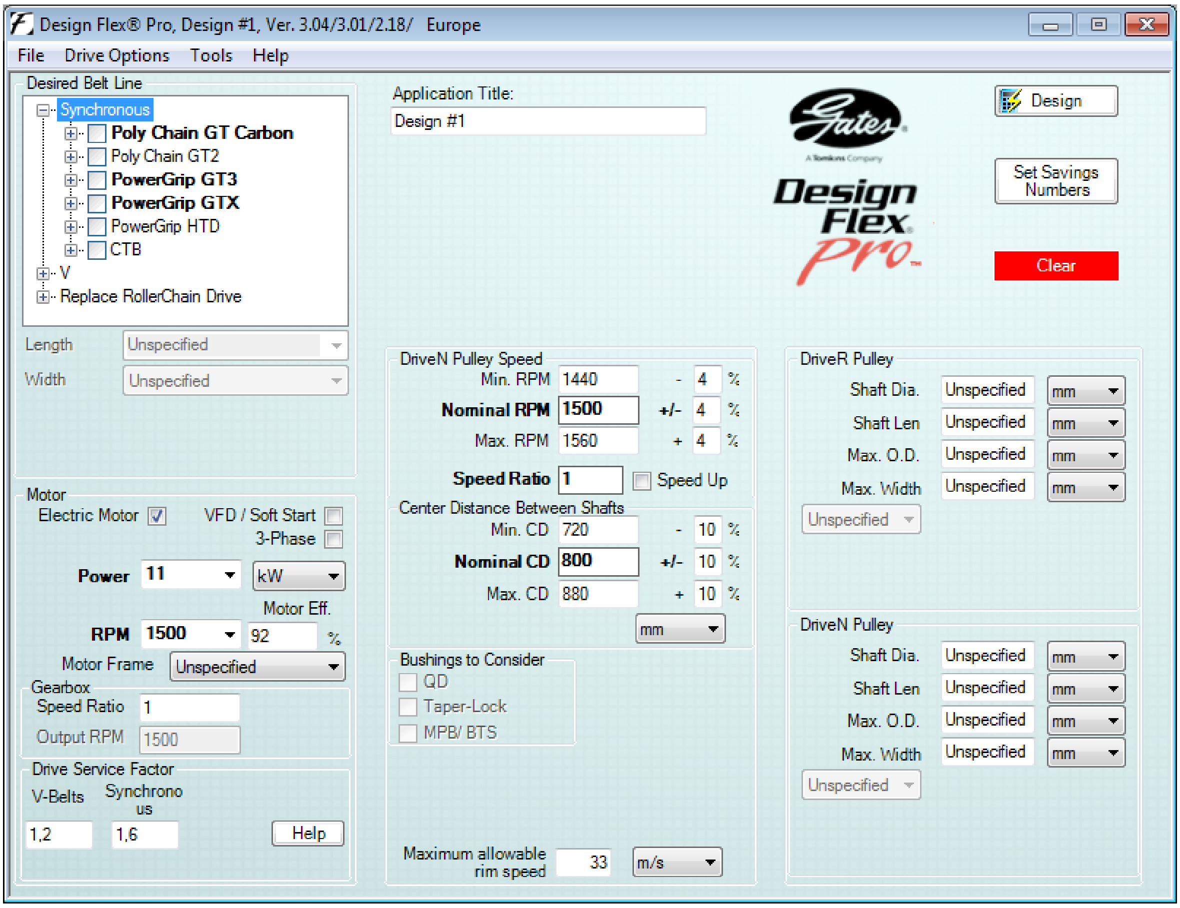


Figure 34. Gates Design Flex Pro -belt drive design software parameter input window.

The design software usually gives a number of different alternatives for the belt drive in the order of best suitable option first. In some cases the belt drive design software gives just one or no solution. In these cases rethinking of power transmission technology might be necessary. Selecting one of the given alternative solutions returns a drive detail report. The drive detail report tells all necessary information such as input parameters, selected drive information including belt and pulley data, tensions information and notes. A drive detail report example is shown in figure 35.

Industrial Belt Design - Drive Detail Report

Design Flex® Pro by the Gates Corporation

Designed For:

Provided By: Jan Liljestrom
Aalto University

Application: **Design #1**

INPUT

Drive Information			DriveR	DriveN
Speed Ratio:	3,00 Down	RPM:	4000,0	1333,3 +/-4%
Input Load:	40 kW, Efficiency: 100,00 %	Maximum Rim Speed:	33 m/s	33 m/s
Service Factor:	1,6	Max Diameter:	Unspecified	230 mm
Design Power:	64 kW	Bushings Checked:	TL, MPB	
Center Distance:	351 to 429 mm	Belts Checked:	Poly Chain GT Carbon	

SELECTED DRIVE

			Belt	DriveR	DriveN
Belt Type:	**Poly Chain GT Carbon - 8MGT**	Part No:	8MGTC-1200-62	22 Grooves	8M-64S-62
Speed Ratio:	**2,91 Down**	Product No:	9274-03150	Non-Stock Item	7726-24064
dN RPM:	**1375,0**	Top Width:	--	--	72,31 mm
Rated Load:	**85,66 kW**	Weight:	350 g	--	7,3 kg
Belt Pull:	**4370 N**	Rim/Belt Speed:	11,7 m/s	11,4 m/s	11,6 m/s
Center Distance:	424,6 mm	RPM:	586,6	4000,0	1375,0
Install/Take-Up Range:	388,8 mm to 425,6 mm	Bushing Part No:	--	--	2517
		Bore:	--	--	12,7 mm - 68,3 mm
		Pitch Diameter:	--	56,02 mm	163,0 mm

TENSION

	New Belt	Used Belt
Rib/Strand Deflection Distance:	6,58 mm	6,58 mm
Rib/Strand Deflection Force:	18 to 19 kg	14 to 15 kg
Sonic Tension Meter		
Belt Frequency:	106 to 110 Hz	90 to 96 Hz

505C/507C Model STM Settings: Mass 4,7g/m,Width: 62 mm/#R, Span: 421 mm

When planning to re-install used belts, measure and record the tension before removing and re-install at the recorded tension.

NOTES

- The DriveR pulley is a special order item. Bore and bush are not known.
- Design Flex Drive Solutions assume Gates products and are not applicable to non-Gates products.
- products are not intended for use in any application where the failure of the product to perform could cause injury or death. This includes use on aircraft propeller or rotor drive systems or other in-flight systems necessary for safe flight.

Figure 35. Gates Design Flex -drive detail report.

One thing to notice in the drive detail report is that the given center distance is the center distance between the two pulleys, when the belt is tensioned properly. The center distance is not the same for a stress-free belt. Another thing to notice is that the proper tension for the belt is given only in frequency and not in force. This means that an expensive frequency measuring device is needed.

Goodyear

More than one belt design software was used in this study. The other belt drive design software used in this study, Goodyear's Maximizer Pro, has a screenshot of its parameter input window presented in figure 36. This free software was downloaded from Goodyear Engineered products internet page (Goodyear engineered products, 2014). The user interface is simpler and easier to use than the Gates software presented in figure 34. In Maximizer Pro the user can see a visualization of the drive layout of the belt drive on the right side of the parameter input window as seen on right side of figure 36. This drive layout is updated in real-time every time the input parameters are changed.

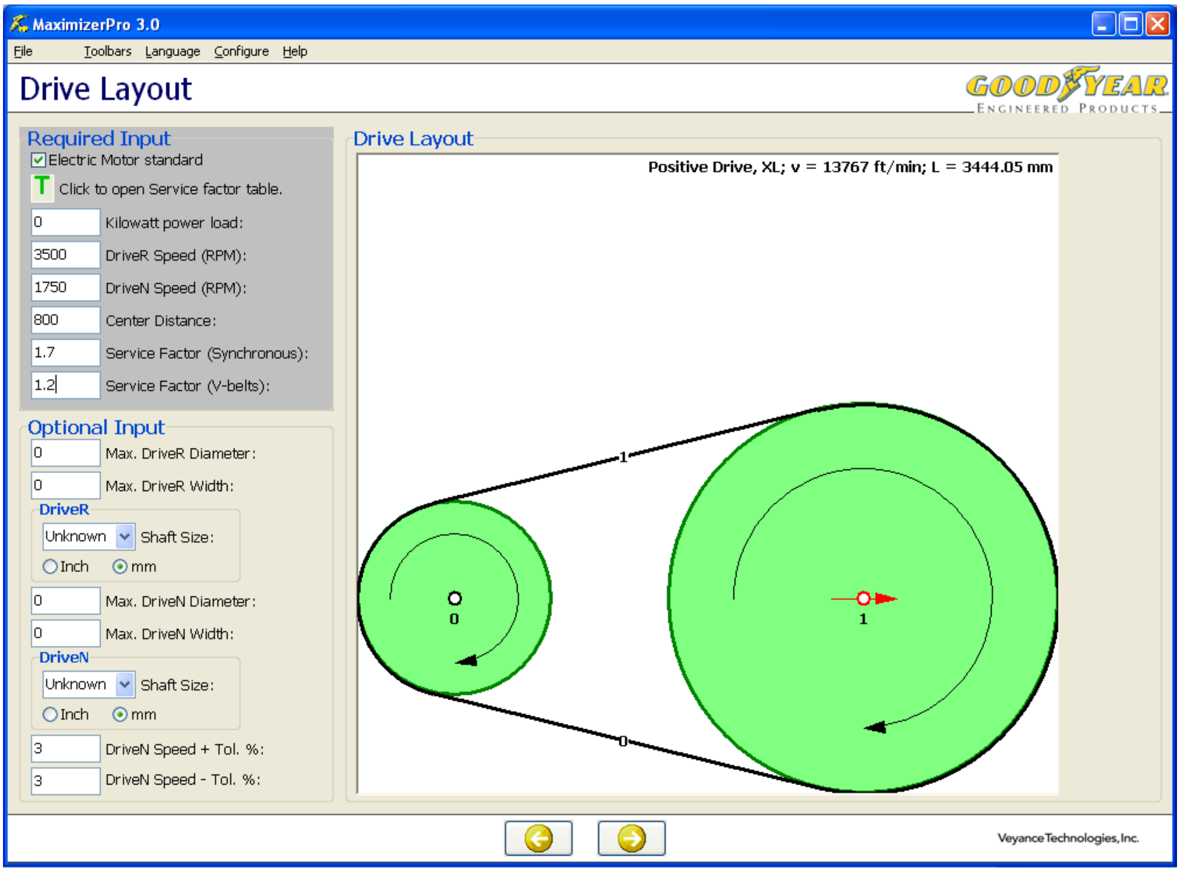


Figure 36. Goodyear's Maximizer Pro 3 -belt drive design software parameter input window.

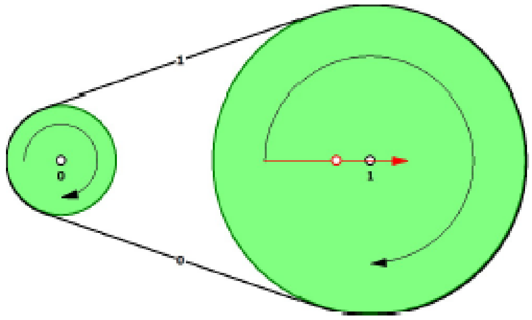
Both Gates and Goodyear's design software work in the same way when giving several solutions for the belt drive for the desired input parameters. Both software products also arrange the results in a reasonable order. In figure 37 the Goodyear Maximizer Pro drive detail report is shown. In addition to the drive detail report by Gates, Goodyear gives additional information such as belt noise and annual energy costs. In using the software by Goodyear I observed that the software leads you to more expensive products.

MaximizerPro Drive Selection Analysis Program — GOODYEAR Engineered Products, Veyance Technologies, Inc.

Belt — Eagle NRG

Belt part number: P 1200
Belt speed; Mass: 26,7 m/s; 0,283 kg/m
Belt power rating: 114,8 kW
Center Distance: 280,9 mm
Belt width, Pitch: 64,0 mm; 8 mm
Belt noise: 87 dB(A)
Required; Actual service factor: 1,7; 1,7
Annual Energy Used; Energy Costs: 36601; 0,06 €/kWh

Belt drive layout



Pulley

Pulley	X [mm]	Y [mm]	Teeth	Diameter [mm]	Flange diameter [mm]	Speed [revs/min]	Width [mm]	Weight [kg]
0	0,0	0,0	40	101,9	/	5000,0	85,0	4,9
1	280,9	0,0	112	285,2	/	1785,7	85,0	13,5

Pulley	Sprocket part number & bushing	Shaft size [mm]/[mm]	Absorbed power [kW]	Required service factor	Side	Type
0	P 40 S-MPB	N/S	67,0	N/A	Inside	Fixed
1	P 112 S-MPB	N/S	67,0	1,7	Inside	Fixed

Hub loads & tensioning

Span	Dynamic hub load (used) [N]	Static hub load (new; used) [N]	Deflection force (new; used) [N]	Deflection [mm]	Span length [mm]	Frequency (new;used) [Hz]
0	3121,6	4749,7; 3392,7	168,0; 123,2	4,1	265,5	177,4; 149,9
1	3121,6	4749,7; 3392,7	168,0; 123,2	4,1	265,5	177,4; 149,9

Installation strand tension

Installation strand tension (new, used): 2512,4 N 1794,6 N

Comments

Recommended centre distance movement for installation and take up: -10,1 mm 3,0 mm
Check shaft lengths!
Ensure Driver and Driven bearings can handle hub load!
Drive must be properly aligned to give intended service life!
Noise levels over 85(dB(A)), may need noise protection for personnel!
The belt and sprocket may not be a stock item. Contact your sales representative for availability and lead time!

Figure 37. Goodyear's Maximizer Pro –drive detail report.

Both Goodyear and Gates belt drive design software products are made to service customers that only need to design a simple belt drive with just the driver and the driven pulley. This means that belt tensioning other than by pulling the two pulleys apart from each other is not considered in these belt drive design software products. This is a challenge when designing a belt drive to NRMM where this kind of belt tensioning might not be possible. The belt tensioning challenge is further considered in chapter 5.2.3.

Goodyear belt drive design software includes design of multi pulley drives. This is good when there is a need for designing a belt drive system similar to a vehicles belt drive for auxiliary devices. However, this part of the software was not suitable for designing two-pulley belt drives with idling tensioner pulleys.

Continental

Both belt drive design software products introduced above turned out to be surprisingly useful for the study done in this thesis. One additional belt drive design software was tried but it turned out not to be suitable for this study. This free software is Continentals Contitech Transmission Designer 7.2 and it was downloaded from the Continental internet page (Continental, 2014). This belt drive design software is significantly different from the ones introduced earlier.

Figure 38. Continental Contitech Power Transmission Designer 7.2 user interface.

The first two belt drive design software products ask few important parameters and have several options from which to choose. When a number of inputs are given, the software gives all suitable solutions that are able to fulfill the requirements. If there are too many solutions, the inputs can be specified more. The Continental belt drive design software does not give alternative solutions for given parameters. It asks the user to give all parameters and then tells if the solution given by the user works or not. By using this program it will probably take longer time to find the best solution when there is a wide selection of different inputs starting from belt nature, type, profile, pitch etc. If the designer has experience and knowledge of the different solutions beforehand, this program can be valuable when optimizing a solution to an end customer's needs. For these reasons, this software was not used in the study.

The user interface of the belt drive design software by Continental is presented in figure 38. The input fields for the belt, geometry and power parameters can be seen on the top of the figure and the results are given in the "results"-field seen at the bottom of the figure. The results field is not as extensive as for the other software introduced in this chapter. The software tells in practice if the given parameters can form a solution or not.

5.2.3 Belt tensioning

When designing the belt drive in the design case of this thesis the belt tensioning appeared to be the most challenging part. This part of the design process was the most time consuming. The belt tensioning issue was the most serious problem in the design case. Almost all belt drives have been done by pulling the pulleys apart from each other but this solution does not fit in this case so there was a need of an external tensioner. Finally tensioning issues were solved by idling tensioner pulleys. Belt tensioning pulleys resulted in a need of higher strength belts. Therefore, carbon fiber reinforced belts was an advantage.

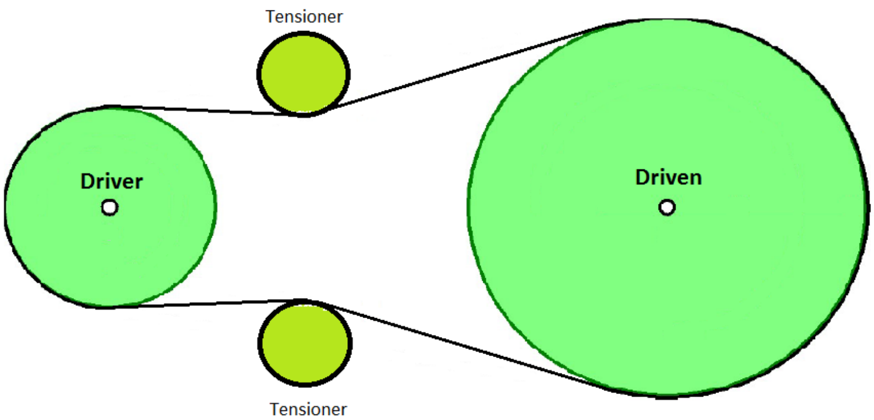


Figure 39. Belt tensioning with double belt tensioners on outside of the belt.

The tensioning of a belt drive, according to most of the retail dealers, is done by distancing the two pulleys from each other. This means that the pulley that is attached to the electric motor should be adjusted. This is done by mounting the motor on rails. The motor is moved in the direction of the rails with a screw so that the belt gets tightened. However, this kind of belt tensioning solution is almost impossible for NRMM in many cases. As for many other NRMM, in the underground mining loader the motor is mounted on a fixed position. The varying load and torque effected on the body requires the belt drive to have a rigid structure keeping the driver and driven axle a certain distance from each other. This results in a fixed axle center distance.

In case of fixed axles the only way of tensioning the belt is by additional tensioner pulleys. However, there are several challenges concerning this. There are two ways of mounting the belt tensioners. The first is to mount them on the outside of the belt as seen in figure 39. In this figure there is one pulley on each side. In the most common cases where the belt drive rotates only in one direction the tensioner is mounted on the loose side. However, in this case study, the belt drive rotates both directions and therefore needs tensioner pulleys on both sides. The advantage of having the belt tensioners mounted on the outside of the belt is that the belt to pulley contact area is wide as can be seen in figure 39. In case of synchronous belt the disadvantage of having the tensioner pulleys on the outside of the belt is that there is a risk of slipping on high rotational speeds. This happens because the outer side of the belt is flat and therefore the pulley surface is also flat and can´t provide enough grip. The slipping is unwanted due to heating and wear of the belt. Outside tensioner pulleys are significantly cheaper due to the flat surface.

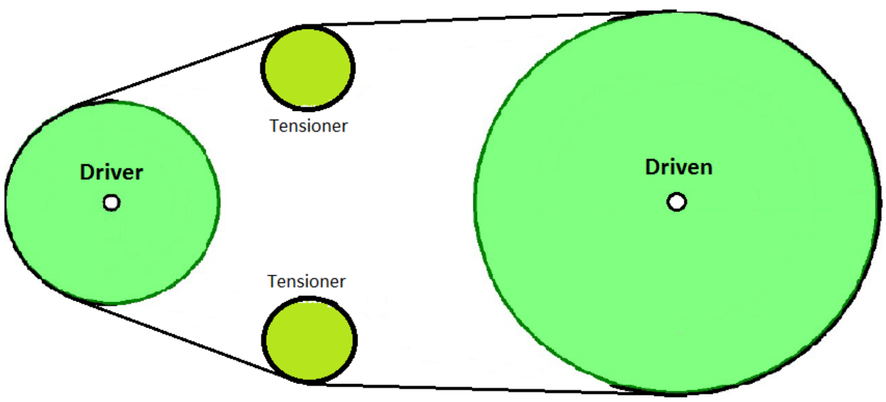


Figure 40. Belt tensioning with double belt tensioners on inside of the belt.

The other way of mounting the belt tensioner pulleys is on the inside of the belt as seen in figure 40. The advantage of this setup is that the tensioner pulleys are grooved when the synchronous belt is grooved on the inside. This allows the belt drive to run at higher rotational speeds without the risk of the belt tensioner pulleys slipping. The disadvantage of having the tensioner pulleys on the inside is the belt to pulley contact for both driver and driven pulley. This contact path is shorter than for having the tensioner pulleys on the outside and there is a risk for slipping especially for the driver pulley belt contact as seen in figure 40.

One additional challenge with belts is that the more times or the tougher the belt has to bend over its path the more fatigue will be caused in the belt. This becomes a greater issue when having the belt tensioners on the outside of the belt but can cause problems also when having the tensioner pulleys on the inside of the belt if the diameter of the tensioner pulley is too small. Carbon fiber reinforced belts gain less fatigue in this kind of tensioning than other belts.

In the design phase an adjustable interface plate for attaching driver and driven axle was invented. This adjustable interface plate is presented in figure 41. The purpose of this interface plate is to have two screws going through it and by rotating these screws the center distance is adjusted. This way the belt could be mounted on the pulleys, after which the belt would be tensioned properly. When the belt has reached its proper tension the interface plate would be mounted on the electric machine and driven axle interface. This version of the design was discarded later due to the uncertainty of the center distance at the proper level of belt tension. This structure was also too complicated to be cost effective to manufacture even for demonstration purposes.

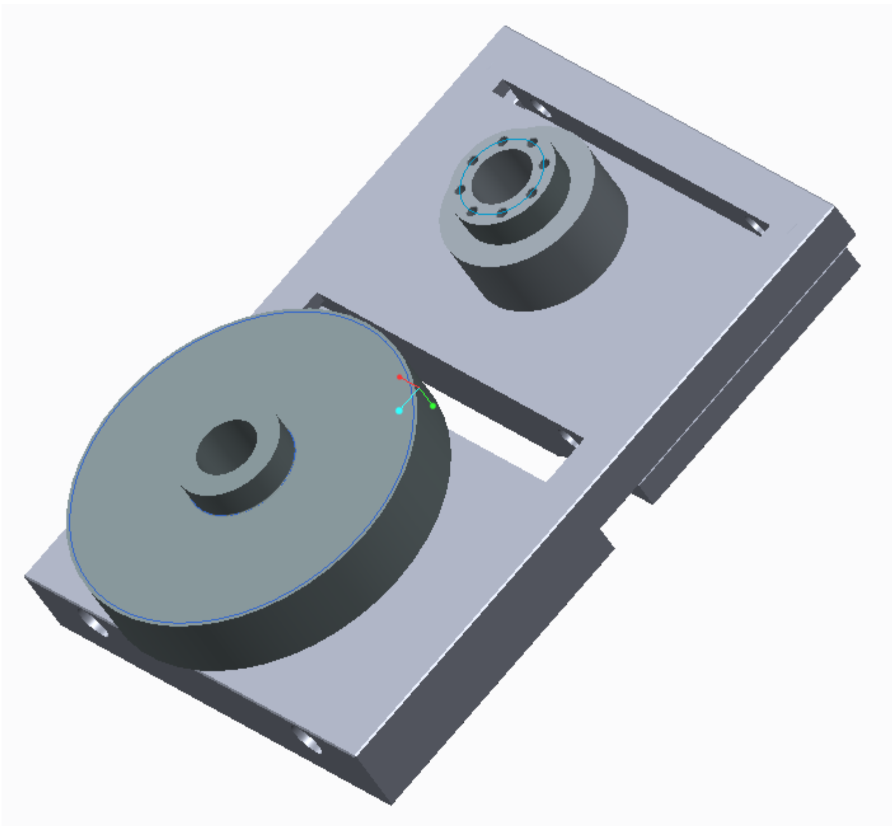


Figure 41. Second version of belt drive design with adjustable interface.

The belt tensioning challenge was solved by the tensioner pulley presented in figure 42. This idler pulley is eccentrically mounted. By mounting the pulley in different positions the belt tension can be varied. The diameter of the tensioner pulley is 60mm. For tensioner pulleys the greater the diameter the less strain is put on the belt because of the gentler curve the belt has to bend around the tensioner.

Figure 42. Idler belt tensioner pulley with eccentric mounting and flat surface.

The type of belt tensioners presented in figure 42 does not have any automatic spring loaded tensioning varying over the driving cycle but instead the set point of the tensioner pulley is rigid. Therefore the belt tension can vary over the driving cycle at different loads but the variation in the tension force was estimated to be significant. The control of the proper value of the belt tension is challenging with the eccentric mounted belt tensioner but this technology was the best found on the market for our application. Therefore belt tension control was also considered as a minor disadvantage.

5.2.4 Mechanical design

The mechanical design phase was the challenging part of this thesis. This was mainly due to the strict requirements in performance of the belt drives. Also the belt tensioning challenges described in the previous chapter were surprisingly time consuming in the design phase. The design process was begun by designing the more challenging belt drive, the one to be mounted on the front axle drivetrain. The reason for this was that the solutions achieved in the more demanding situation can be easily implemented in the less demanding case.

In terms of strength and durability, part manufacturer´s guidelines were followed. These guidelines were useful when choosing bearings and shaft sizes as well as when designing mountings for bearings. In terms of material and thickness of material, general machine design guidelines were followed. According to trials and errors of previous studies in the research group, double bearings were considered as a requirement, to be on the safe side. In the first version of the front belt drive design the bearings were designed to be mounted in frame plates on both sides of the belt pulleys as seen in figure 43. This figure shows a cross section of the belt drive cut through both belt pulleys. Double bearings are designed to be for both belt pulleys. One is a locating bearing, taking both axial and radial loads and the other is a floating bearing taking only radial load.

There following challenges appeared for this design:

- Alignment of frame plates
- Assembling
- Belt tensioning
- Maintaining and monitoring

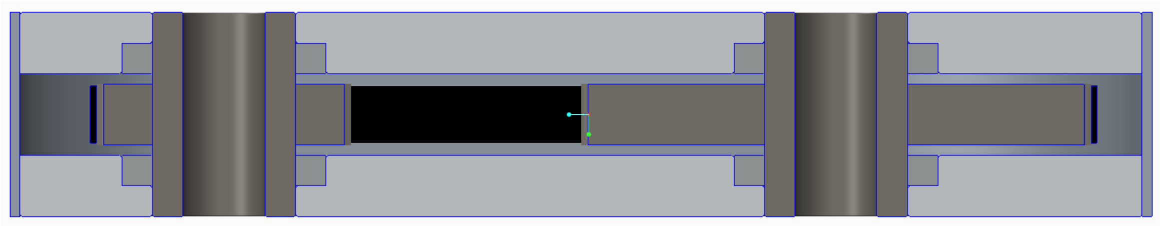


Figure 43. Cross section view of first version of front axle belt drive design.

Because of the many challenges and weaknesses of the first version of the belt drive design, it was discarded. The greatest challenge in the first version of the design was belt tensioning, especially when there is a need for symmetrical torque compatibility in both rotational directions. A second version of the design was an adjustable interface seen in figure 41. In this version of the design the belt tensioning could be performed by moving the different parts away from each other. This is done by turning screws going through an interface plate. However, this design seemed to be too complicated to implement so it was discarded in an early stage. Main challenges were the complexity of the design with several moving parts and uncertainties regarding the axle center distances in the different parts of the belt drive as mentioned in a previous chapter. Version two of the design would also have needed to have some frame structure on the other side of the belt pulleys that is not seen in figure 41. This frame structure would have needed to be adjustable as well as the interface plate.

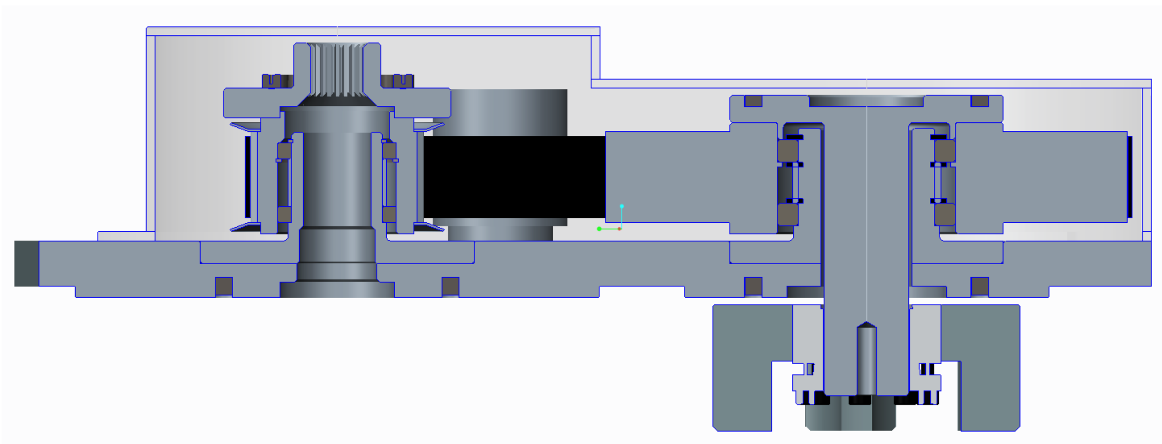


Figure 44. Cross section view of the third version of the front axle belt drive design.

After the trials and errors in design versions one and two, the third design version consisted of only one frame plate. This frame plate was an interface plate to the input and output mountings of the belt drive. At this point studies in belt tensioning had reached the level that idler tensioner pulleys were decided to be implemented in the design. The third and last version needed many revisions and updates before all the parts were possible to be manufactured with the tools available and for the mountings of all necessary parts to be possible. A cross section view of the third and final version of the front axle belt drive is presented in figure 44. This figure shows how the belt pulleys are attached via bearings to hollow shafts with flanges that are mounted to the frame plate. The electric motor interface is on the left side of figure 44. The electric motor is mounted to the frame plate. The motor axle goes through the hollow shaft and is attached to the part with splines on the other side of the belt pulley. This part transmits the torque from the motor to the belt pulley. The splined part can be removed from the belt drive in order to release the electric motor from the rest of the drivetrain. This is needed in the commissioning phase of the electric motor. An additional requirement for the belt drive design was that it also needed to work for the test bench located in Aalto University Automotive Engineering laboratory for performance measurement purposes. This requirement of compatibility to two different interfaces was an additional time consuming challenge in the design phase.

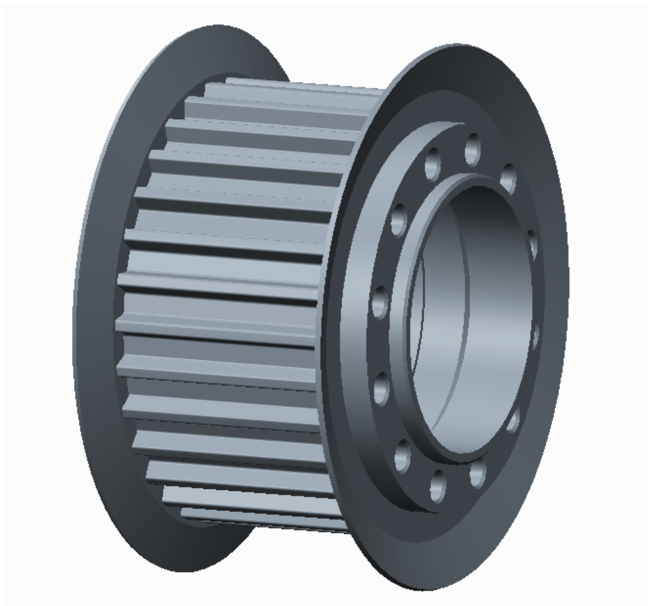


Figure 45. Driver pulley of front axle belt drive.

Figure 45 presents the driver pulley of the front axle belt drive. The belt drive components determined for the front axle drivetrain were the ultimate components that Gates had to provide (Gates, 2014). The parts were ordered from SKS, the local retail dealer of Gates. This local retail dealer was able to provide machining of belt pulleys as a whole according to drawings, which was a great advantage. As seen in figure 45 the pattern of the belt pulley is typical for synchronous belt drive pulleys. This type of pattern needs flanges on both sides of the pulley for the belt not to escape in axial direction in case of axial misalignment.

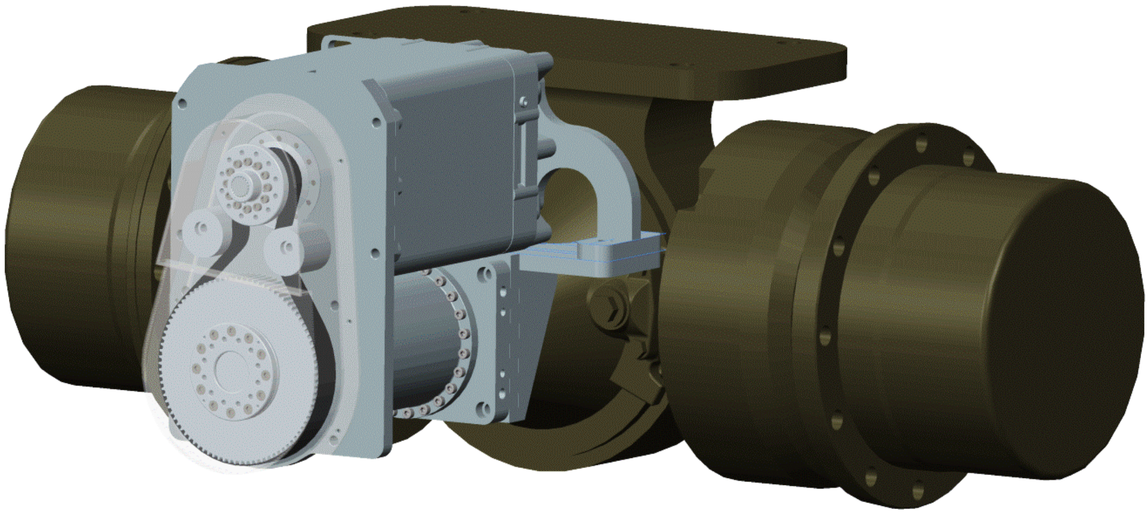


Figure 46. Front axle drivetrain from electric motor to wheel hubs.

The whole drivetrain for front wheels from electric motor to wheel hubs is presented in figure 46. As can be seen in the figure, the belt drive parts are easy to mount and maintain. Also monitoring of belt drive in this demo is easy through the transparent cover made of polycarbonate. The cover is also needed for protecting the surroundings from particles flying from the rotating axles and in case of belt breakdown. The cover was made out of plastic in this case, but if there is a need of cooling it can also be made out of thermal conductive material such as sheet metal. The front axle drivetrain which resulted was surprisingly compact compared to the size of the electric motor but still there seemed not to be many millimeters left on each side in the place where the drivetrain was designed to be mounted.

When all the greatest challenges in the design process of the front axle belt drive were completed, the design of the rear axle drivetrain was significantly easier and less time consuming. The parts chosen for the rear axle belt drive were the ultimate components that Goodyear had to provide (Goodyear, 2014). The parts were ordered from Jens S., the local retail dealer of Goodyear engineered products. This local retail dealer was only able to provide machining of minor modifications to billet pulleys provided by the manufacturer. This was probably caused by the complex shape of the belt pulley pattern seen in figure 47.

Figure 47. Driver pulley of rear axle belt drive.

The belt pulley chosen for the rear axle belt drive presented in figure 47 has a significantly different pattern than traditional synchronous belt pulleys as seen earlier in figure 45. This more complicated pattern is more challenging to manufacture but has several advantages for example higher efficiency, lower noise and higher power capacity at narrower belt widths. In addition, this pattern provides a self-centering belt so there is no need of flanges as in the traditional pattern belt pulley in figure 45.

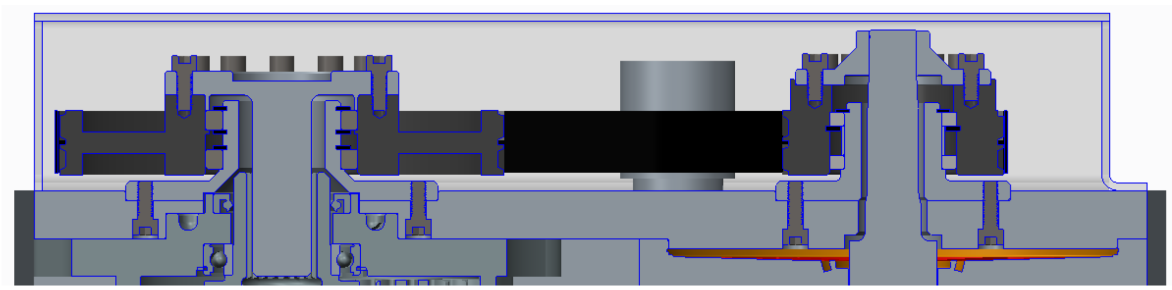


Figure 48. Cross section view of rear axle belt drive design.

Figure 48 presents the cross section view of the rear axle belt drive. As seen in the figure, the belt drive design is similar to the one on the front axle drivetrain presented earlier in figure 44. The electric motor interfaces are similar for both belt drives, but the rear axle belt drive is attached to a gearbox which complicates the design. For both front and rear belt drives the idler belt tensioner pulleys are wider than needed as can be seen in the figures. This is because there were no narrower tensioners of the same kind available at the market at this time. In addition, wider pulleys do not limit the performance of the belt drive.

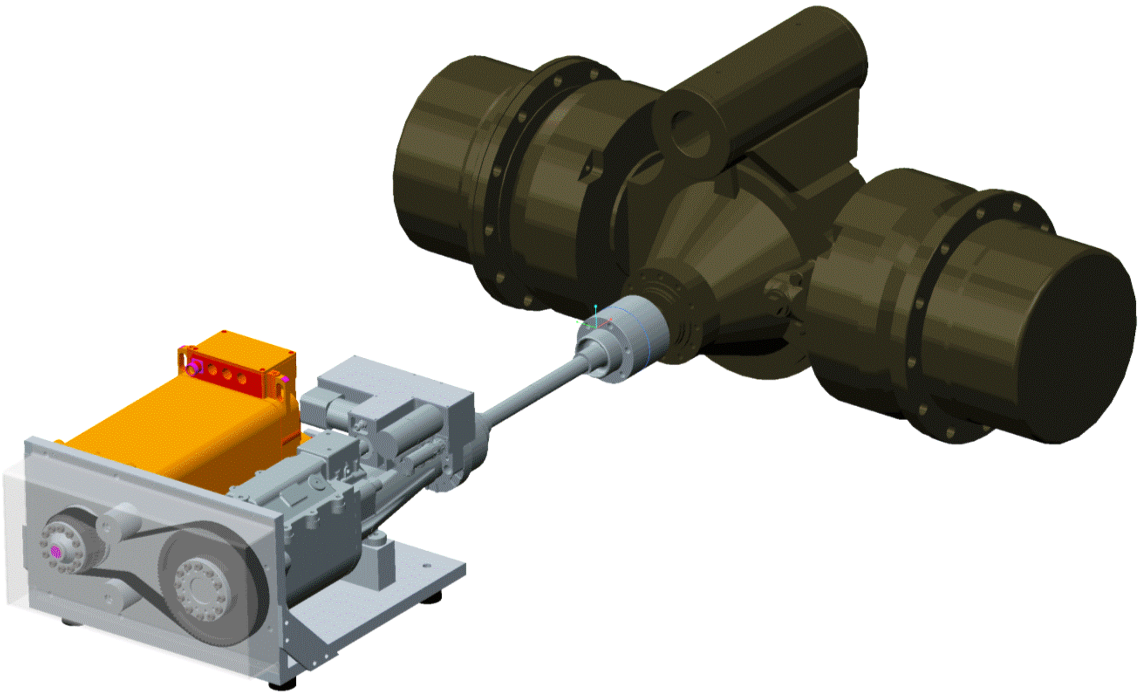


Figure 49. Rear axle drivetrain from electric motor to wheel hubs.

The whole drive train for the rear axle from electric motor to wheel hubs is presented in figure 49. This drivetrain as a whole was easier to design because a lot of experience was gained from the design of the drivetrain for the front axle. There was also more space available for the rear axle drivetrain than for the front one. Therefore, less work on optimization of structures was needed to be able to fit the belt drive in the back carriage. The front axle of the mining loader is rigid but the rear axle is of oscillating type as seen from the upper mounting in figure 49 right side. The long axle going from transmission to differential seen in the figure has flexible lattice joints in both ends which allow the change of the position of the differential when the rear axle is rocking. More pictures and 3D-drawings of the designed components and assemblies are presented in appendix 1.

5.3 Implementation

The implementation part of the thesis was to facilitate ordering of parts for the belt drives. Assembling and testing of the belt drives are performed after this thesis. At the time of finishing this thesis all parts included in the belt drives and the rest of the drivetrains were ordered and most of them arrived. In figure 50 there is presented the parts for the front axle belt drive. The mechanical drawings for these machined parts as well as the exploded views of the assemblies are shown in appendix 1 on pages 68 to 69 and 71 to 75.

Figure 50. Parts for the front axle belt drive.

Figure 51 presents the parts for the rear axle belt drive. The mechanical drawings for these machined parts as well as the exploded views of the assemblies are shown in appendix 1 on pages 68, 70 and 75 to 78.

Figure 51. Parts for the rear axle belt drive.

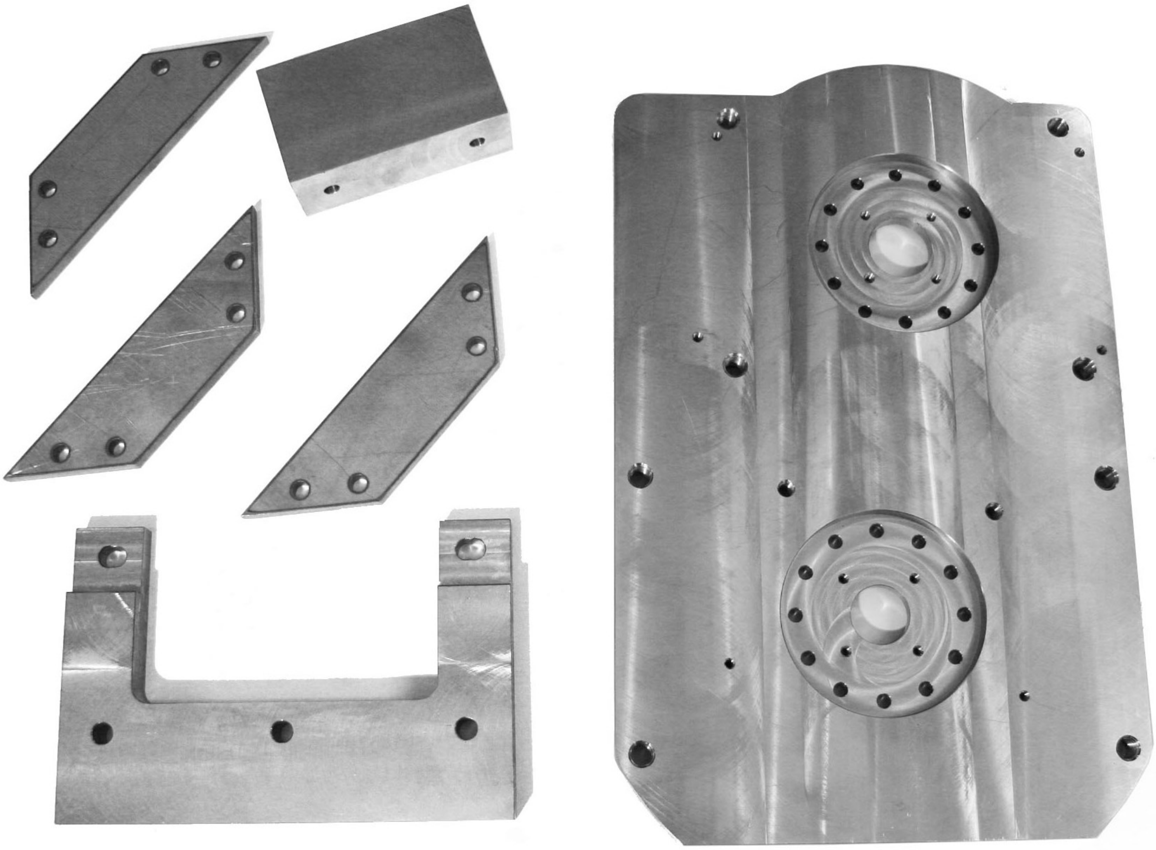


Figure 52. Rear axle drivetrain components on left side and the front axle belt drive interface plate on right side.

Figure 52 presents rear axle drivetrain components on the left side and the front axle belt drive interface plate on right side. The mechanical drawings for the rear axle drivetrain components presented in the figure are shown in appendix 1 on pages 80 to 81. The mechanical drawing for the front axle belt drive interface plate is found in appendix 1 on page 79.

A special tool is needed for assembling the bearings in the belt drives designed in this thesis. This unique tool for bearing assembly was designed and manufactured within this thesis work. Figure 53 presents this tool. The mechanical drawings for the parts as well as the exploded view of this tool are shown in appendix 1 on pages 83 to 85.

Figure 53. Tool for bearing assembly.

5.4 Discussion and further studies

The case study of this thesis was the most rewarding part taking up to four fifths of the allocated 6 months' time. Some constraints were challenging to fulfill for example the limitation in volume for the front belt drive. This could have possibly been easier reengineering the whole front axle. However, this would have been out of scope for this thesis.

Implementing double bearing in a separate belt drive module was as a single matter a quite demanding task. Usually belt pulleys are mounted straight on the electric motor axle. Some electrical motors have bearings that are able to carry the load of a belt drive straight mounted on the motor axle. However, not all motors allow this.

The belt drive calculation software products were demanding to handle but it is more a matter of experience of the technology and especially belt manufacturer product range. If the belt drives designed would have been tensioned by pulling apart the driver and driven axle, this would have been much easier from belt point of view. The belt drive design software products were able to take only this kind of tensioning into account. Some of the belt drive design software products needed a lot of information before they were able to prove if the belt drive would work or not. The belt drive design software products would have worked better for this thesis if they would have suggested more possible solutions.

The case study was successful. As mentioned earlier when the case study was begun there were many uncertainties considering the belt drive and this case study was meant to be a test to try to implement a belt drive in a case like this. The belt drive design was performed successfully and many great challenges were completed during this time. At no time was the original goal of implementing a belt drive abandoned. As a whole the case study was successful and gave new information and brought new skills to the performer of the case study himself and to the company involved in the case study.

Results of this study indicate that belt drives are possible to implement in NRMM. To facilitate the implementation, electrical motor manufacturers could manufacture electrical motors that have built-in bearings that can carry required loads of a belt drive so that belt pulleys can be straight attached to the motor axle. This might also need some re-engineering of electrical motor axles. Anyhow, this is certainly something profitable for electrical motor manufacturers to develop.

For future studies it would be preferable to perform following tests after the belt drives have been assembled:

- Efficiency measurements
- Vibration measurements
- Temperature measurement for belts and bearings
- Noise measurement
- Harshness evaluation
- Wear evaluation

All the tests mentioned above would preferably be performed in a test bench. Temperature measurements can be performed by running both drivetrains, including electric motors and belt drives, against each other and monitoring the temperature with a thermal camera. Belts and bearings are the most sensitive parts. Therefore their temperature should be measured. Temperature measurements should be performed both with and without cover. Testing a cover made of heat conductive material is reasonable.

Harshness evaluation is objective, and can be based on for example looking’s or driving and maintaining comfort.

Belt wear is good to evaluate. This is easiest to perform using a failure analyses guide by belt manufacturers. Belt failure mechanisms are listed in appendix 2. For a more extensive version see Synchronous Belt Failure Analysis Guide (Gates, 2013). Prior to assembling the belt drives, it is advisable to read the synchronous belt guide provided by Gates web page (Gates PowerGrip, 2014). The document can be downloaded from the bottom of the given web page under “E-Z Download” by clicking “PowerGrip® timing belt”.

Conclusion

The aim of this thesis was to introduce and discuss the electromechanical transmission challenges in non-road mobile machinery (NRMM) drivetrains using modern belt drive as the domain. This was performed first by considering the differences between traditional drivetrains in passenger cars and NRMM. After this, hybrid and electric drivetrains were introduced and the transmission challenges of each were considered. Matters considering the design perspective of a hybrid or electric drivetrain were introduced. In the end of the literature research part of the thesis, different types of power transmissions were introduced as solutions for the electromechanical transmission challenge.

The case study part of this thesis was the main target. Therefore the literature part is concise. The literature research idea was to introduce the reader to the characteristics of the hybrid and electric world of drivetrains as well as to give some basic knowledge on the concept level design process.

The case study part of the thesis was performed in the timeframes of the thesis work. The aim of this case study was to design and order the parts for the belt reduction gears on front and rear axle drivetrains of the 14 ton underground mining loader. Gear ratios were 3 for the belt drive on the front axle drivetrain and 2 for the rear. Two commercial belt drive manufacturers´ engineering guide lines were followed and based on these, two belt drives were chosen. The mechanical parts were engineered and drawn to the level that they were able to be manufactured. All parts were ordered and many of them arrived before finalizing the thesis work. The belt drives work in theory according to the manufacturers. Future test drives will show how well the belt drives will manage in practice. Electric motors with bearings carrying both axial and radial load would have made the design task easier. This thesis, however, proved that it is possible within a time frame of 6 months to overcome the challenge of having non-belt drive compatible electric motors for prototype or technology evaluation vehicles.

As a whole it seems, based on engineering document of manufacturers and usage of volume, that belt drives can be used as a first reduction gear from the electric motor to the axle. The thesis includes many explanatory figures to help the reader to estimate the design solution for this case and hopefully to help the reader to evaluate if a belt drive could be one option to be considered when needing a reduction gear between higher speed electric motor and conventional mechanical gear.

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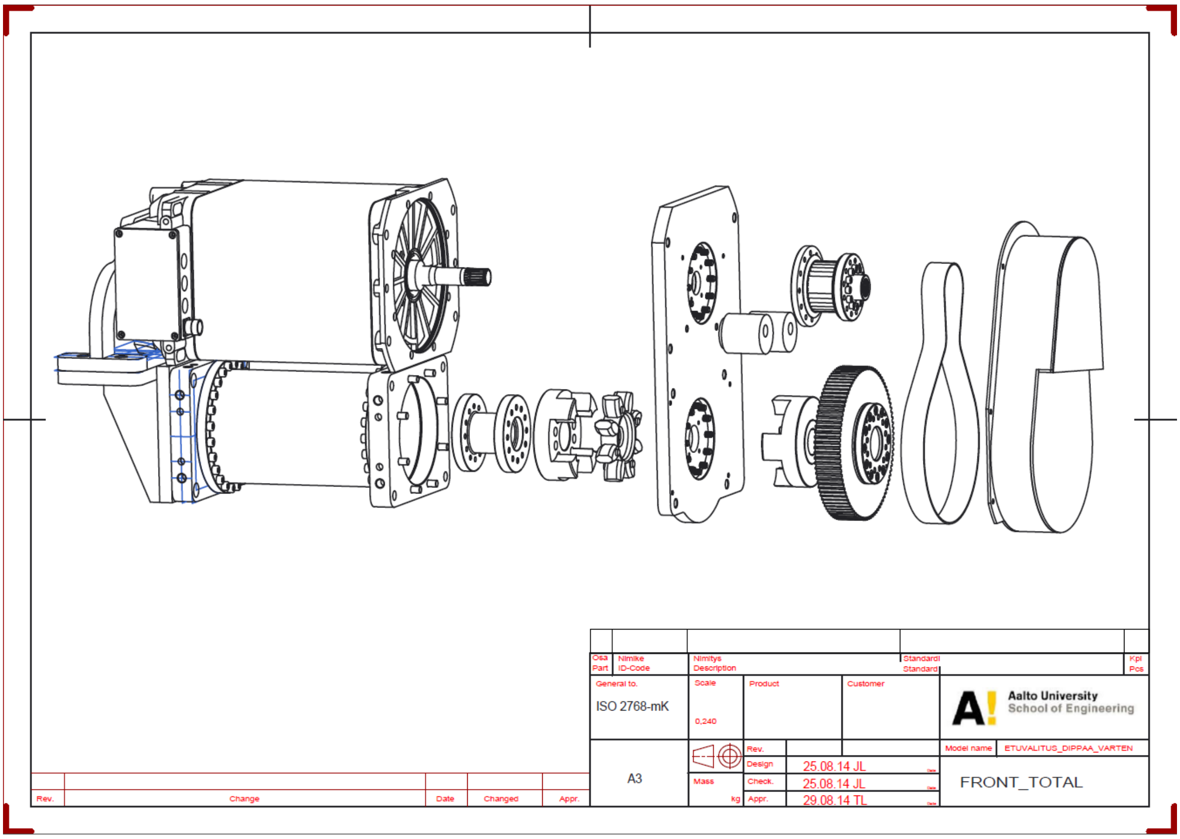
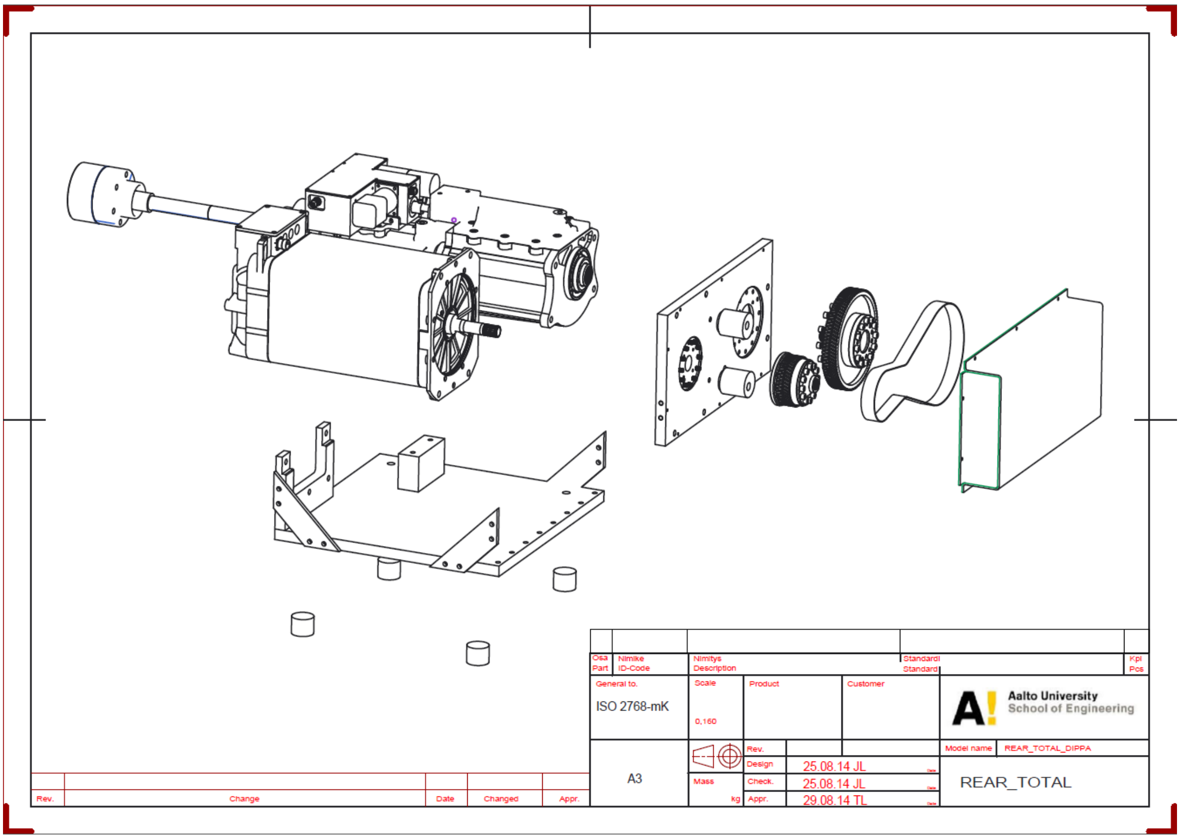
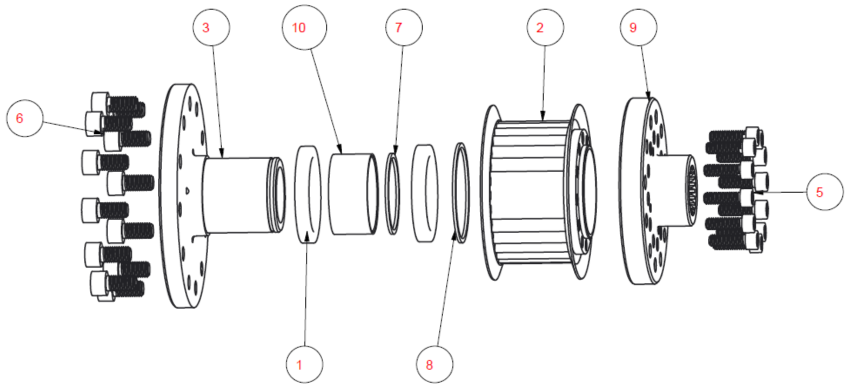
Appendix 1 : Drawings of components and assemblies







ITEM NO	PART NUMBER	QTY
1	BEARING_40_52_7	2
2	ETU_PIENI_HIHNAPYORA4	1
3	ETU_YLA_PUTKI_LAAKEREILLE2	1
4	LAIPPA_ETU_PIENI_HP	2
5	M6_KORKEAK_20	12
6	M8_KORKEAK_16	12
7	SNAP_RING_EXTERNAL_40	1
8	SNAP_RING_INTERNAL_52	1
9	SPLAINIADAPT_ETU_MOTTI_3	1
10	VALIPUTKI_LAAKER_ETUY	1

Osa Part | Nimike ID-Code | Nimitys Description | Standardi Standard | Kpl Pos

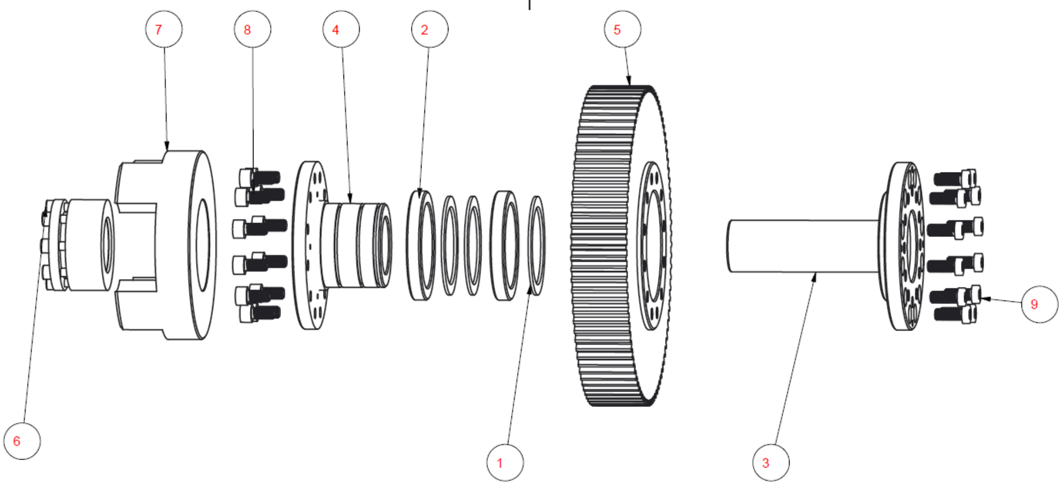
General to. ISO 2768-mK | Scale 0,667 | Product | Customer

Aalto University School of Engineering

A3 | Mass kg | Rev. | Design 25.08.14 JL | Check. 25.08.14 JL | Appr. 29.08.14 TL | Model name ETU_PIENI2

ETU_PIENI2

Rev. | Change | Date | Changed | Appr.



ITEM NO	PART NUMBER	QTY
1	AKSELIVARMISTIN_60	3
2	BEARING_60	2
3	ETU_ALA_AKSELI	1
4	ETU_ALA_PUTKI_LAAKERILLE	1
5	ETU_ISO_HIHNAPYORA	1
6	JENS_KIRISTIN	1
7	KTR_ETU_HIHNAV_ALA	1
8	M8_KORKEAK_16	12
9	M8_MATALAK_20_20	12

Osa Part | Nimike ID-Code | Nimitys Description | Standardi Standard | Kpl Pos

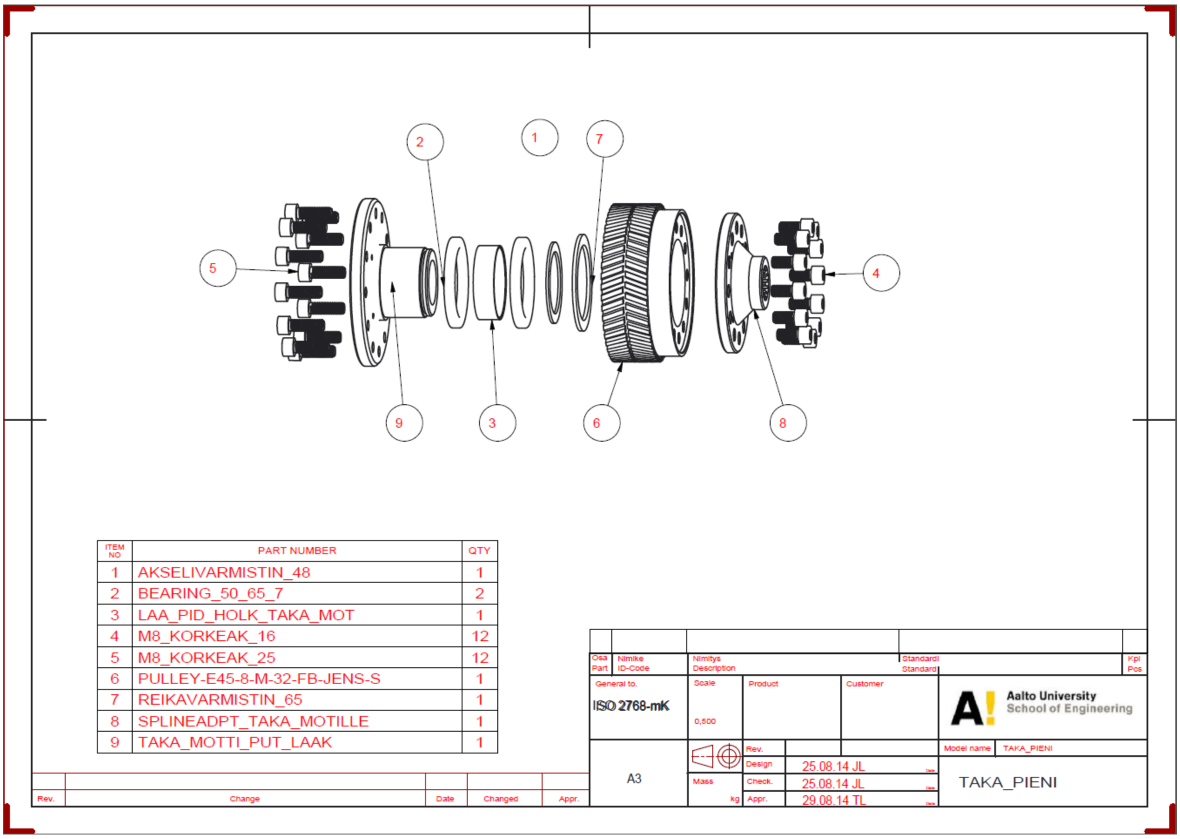
General to. ISO 2768-mK | Scale 0,500 | Product | Customer

Aalto University School of Engineering

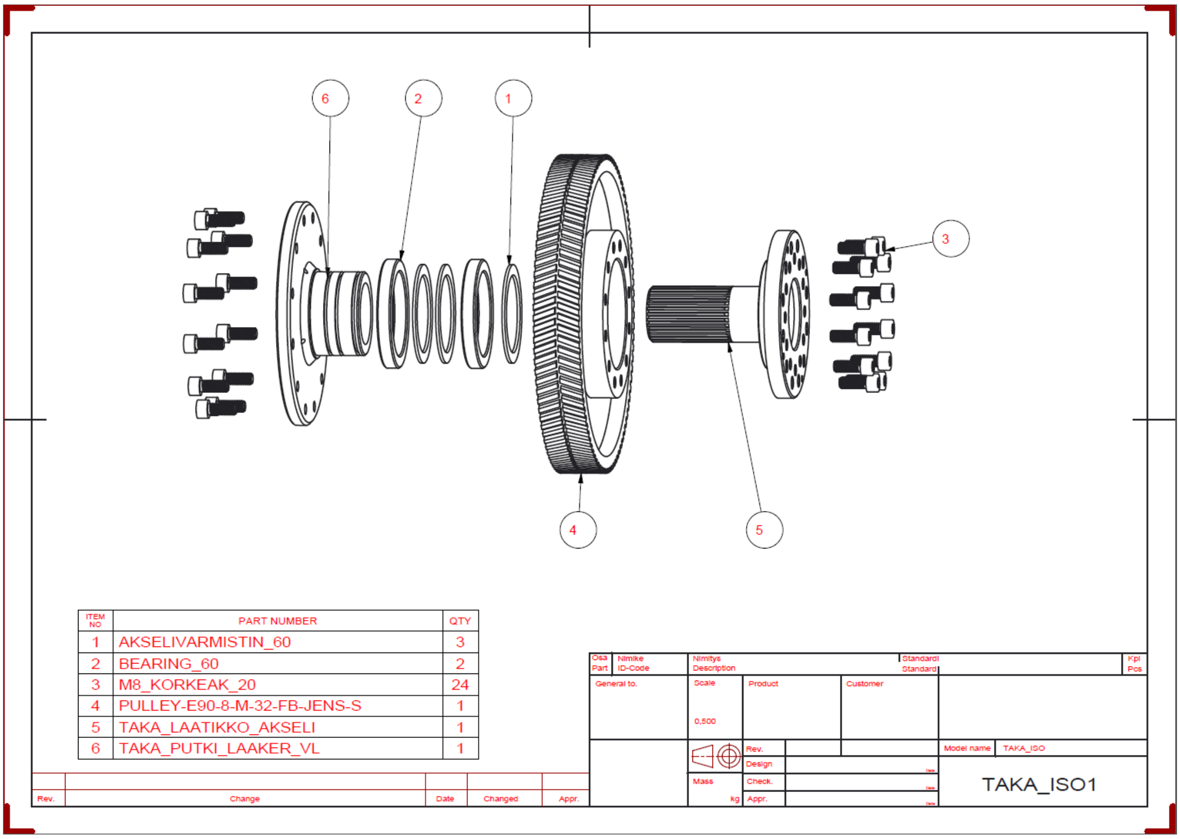
A3 | Mass kg | Rev. | Design 25.08.14 JL | Check. 25.08.14 JL | Appr. 29.08.14 TL | Model name ETU_ISO

ETU_ISO

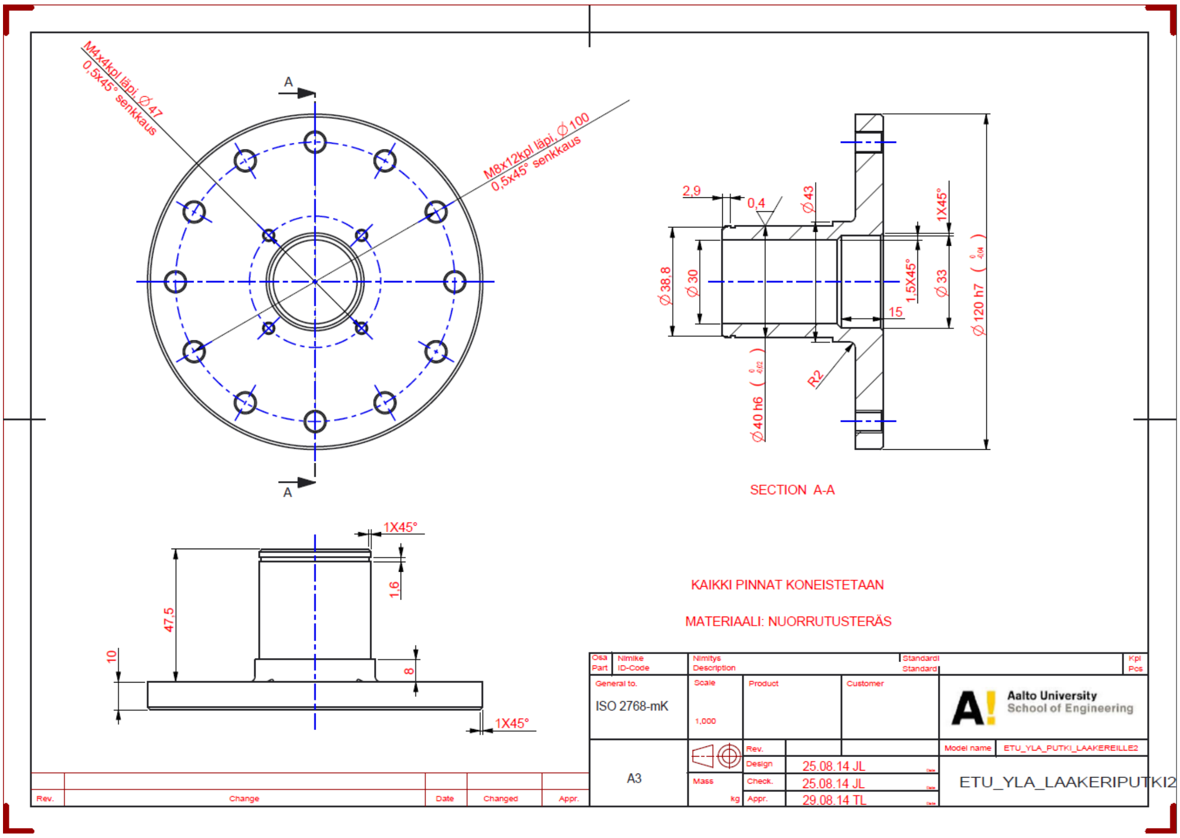
Rev. | Change | Date | Changed | Appr.

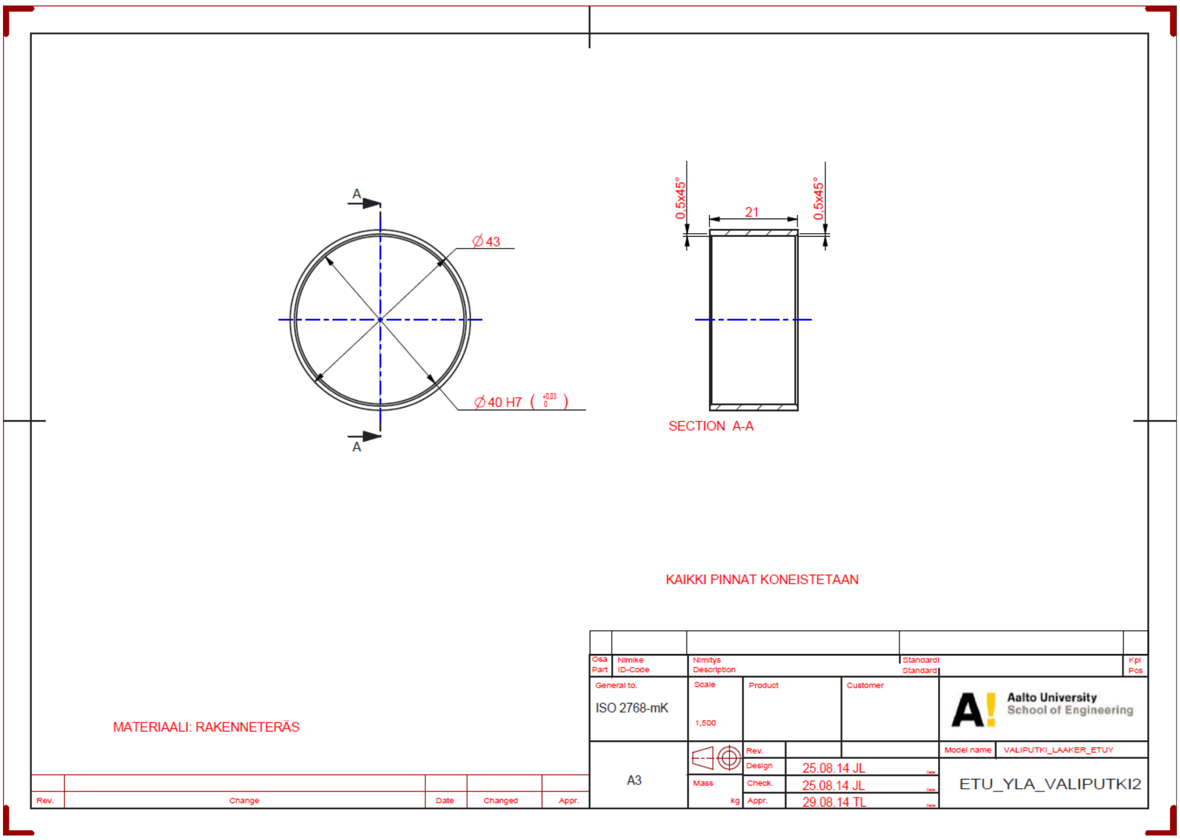


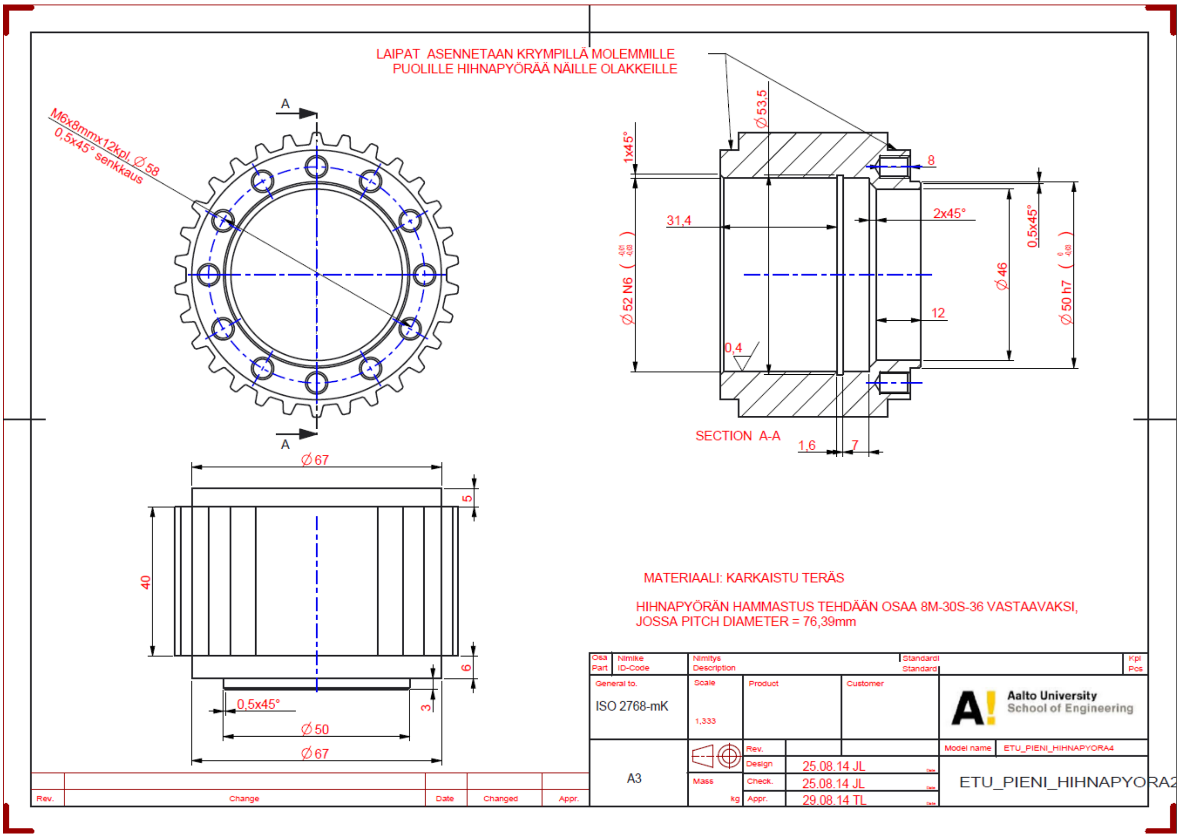
ITEM NO	PART NUMBER	QTY
1	AKSELIVARMISTIN_48	1
2	BEARING_50_65_7	2
3	LAA_PID_HOLK_TAKA_MOT	1
4	M8_KORKEAK_16	12
5	M8_KORKEAK_25	12
6	PULLEY-E45-8-M-32-FB-JENS-S	1
7	REIKAVARMISTIN_65	1
8	SPLINEADPT_TAKA_MOTILLE	1
9	TAKA_MOTTI_PUT_LAAK	1



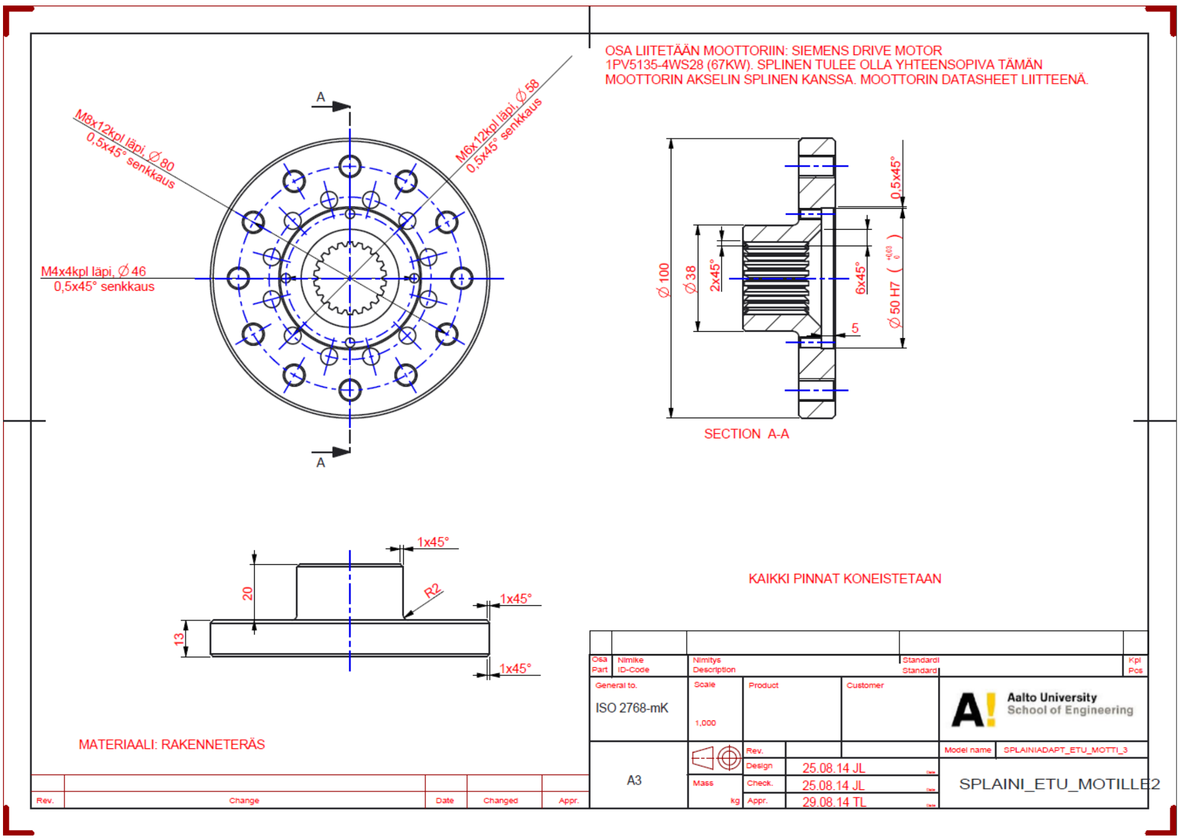
ITEM NO	PART NUMBER	QTY
1	AKSELIVARMISTIN_60	3
2	BEARING_60	2
3	M8_KORKEAK_20	24
4	PULLEY-E90-8-M-32-FB-JENS-S	1
5	TAKA_LAATIKKO_AKSELI	1
6	TAKA_PUTKI_LAAKER_VL	1


M4x4kpl läpi, ∅47
0,5x45° senkkaus
M8x12kpl läpi, ∅100
0,5x45° senkkaus
SECTION A-A
KAIKKI PINNAT KONEISTETAAN
MATERIAALI: NUORRUTUSTERÄS
ISO 2768-mK
A3
Aalto University
School of Engineering
Design 25.08.14 JL
Check. 25.08.14 JL
Appr. 29.08.14 TL
ETU_YLA_LAAKERIPUTKI2

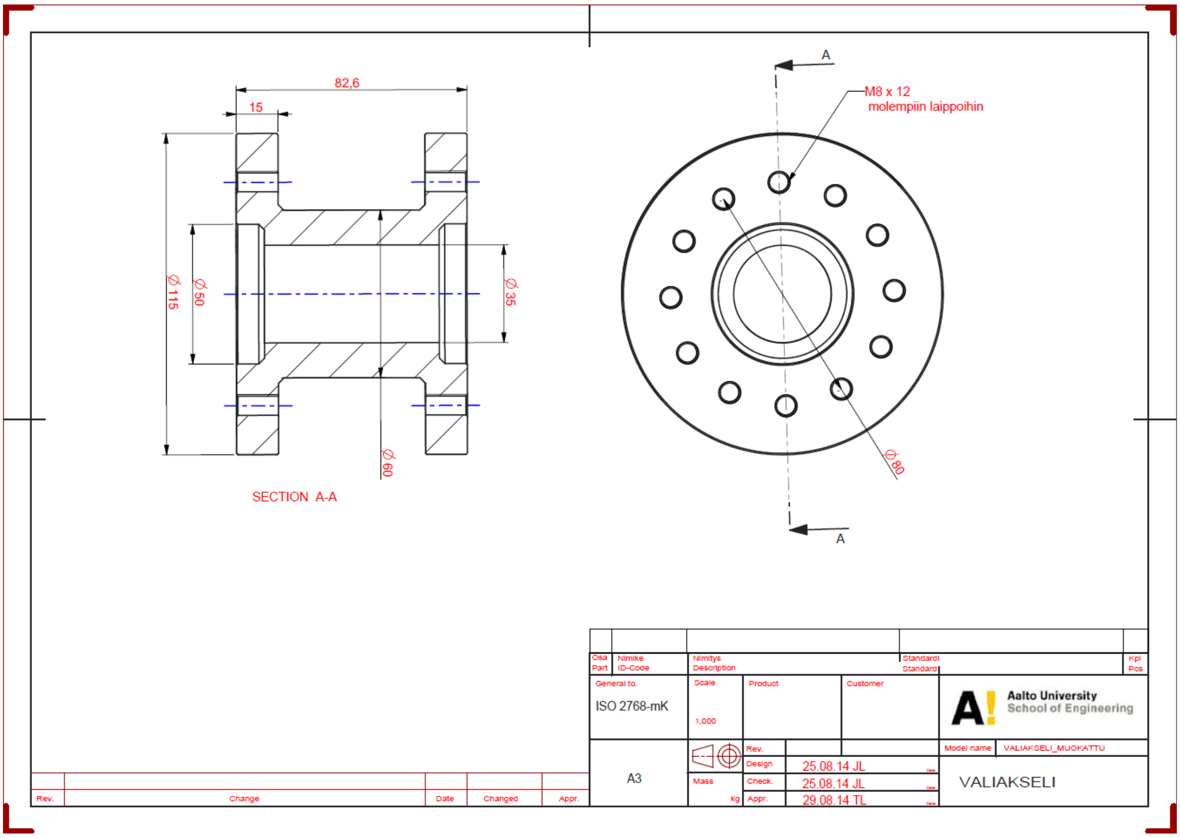

∅43
∅40 H7
SECTION A-A
KAIKKI PINNAT KONEISTETAAN
MATERIAALI: RAKENNETERÄS
ISO 2768-mK
A3
Aalto University
School of Engineering
Design 25.08.14 JL
Check. 25.08.14 JL
Appr. 29.08.14 TL
ETU_YLA_VALIPUTKI2

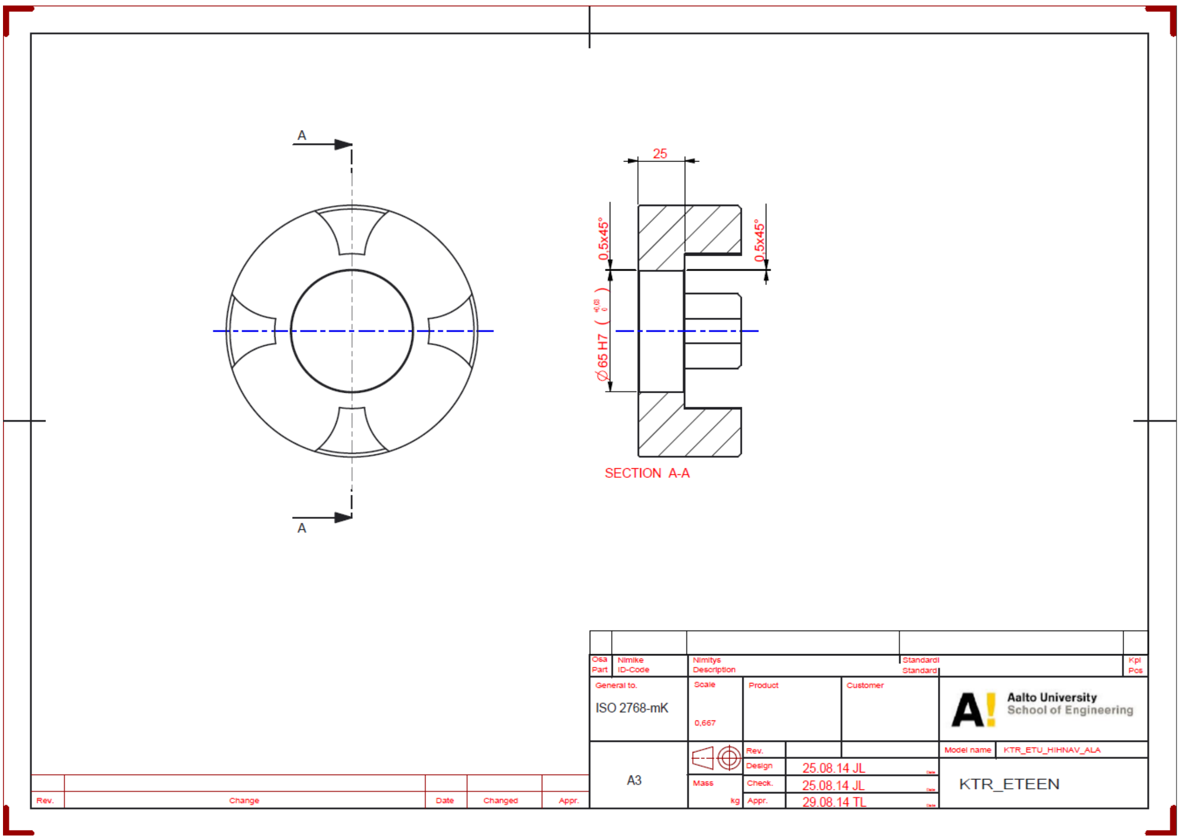


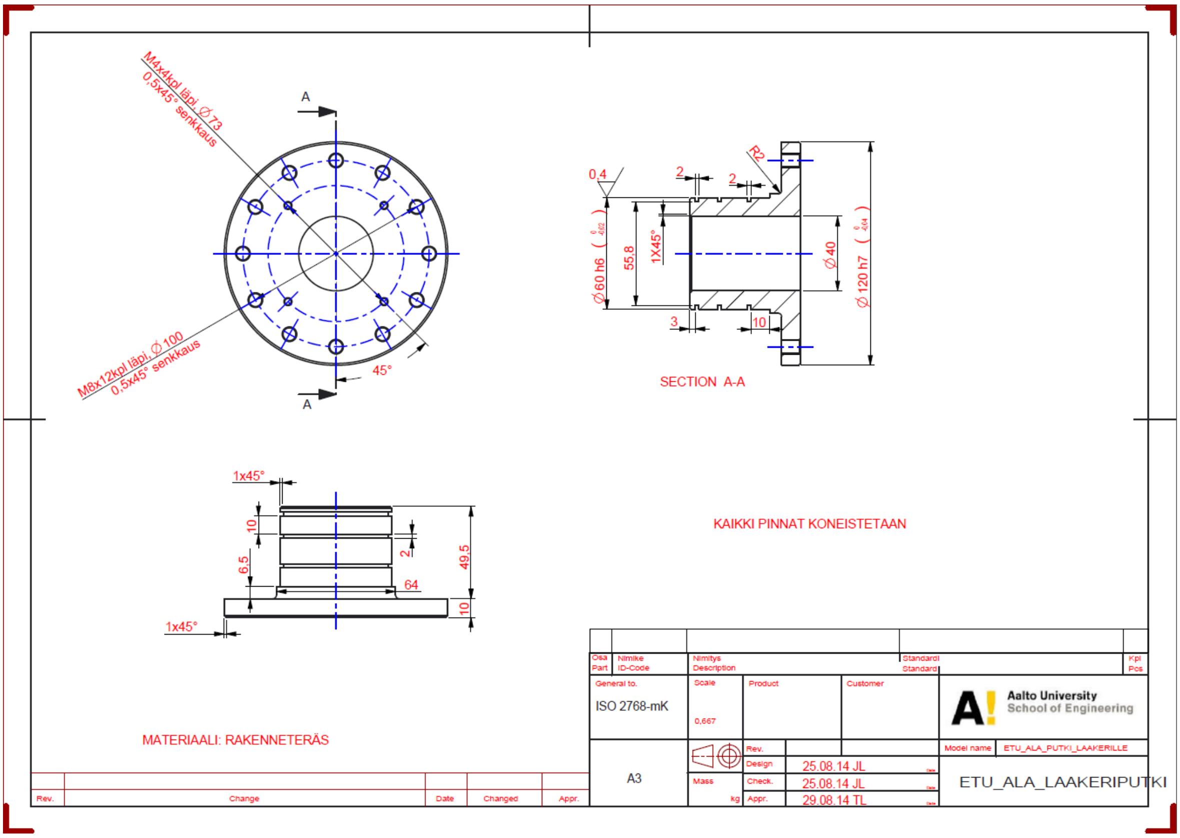
LAIPAT ASENNETAAN KRYMPILLÄ MOLEMMILLE PUOLILLE HIHNAPYÖRÄÄ NÄILLE OLAKKEILLE
M6x8mmx12kpl, Ø 58
0,5x45° senkkaus
SECTION A-A
MATERIAALI: KARKAISTU TERÄS
HIHNAPYÖRÄN HAMMASTUS TEHDÄÄN OSAA 8M-30S-36 VASTAAVAKSI, JOSSA PITCH DIAMETER = 76,39mm
ISO 2768-mK
A3
Aalto University
School of Engineering
ETU_PIENI_HIHNAPYORA2

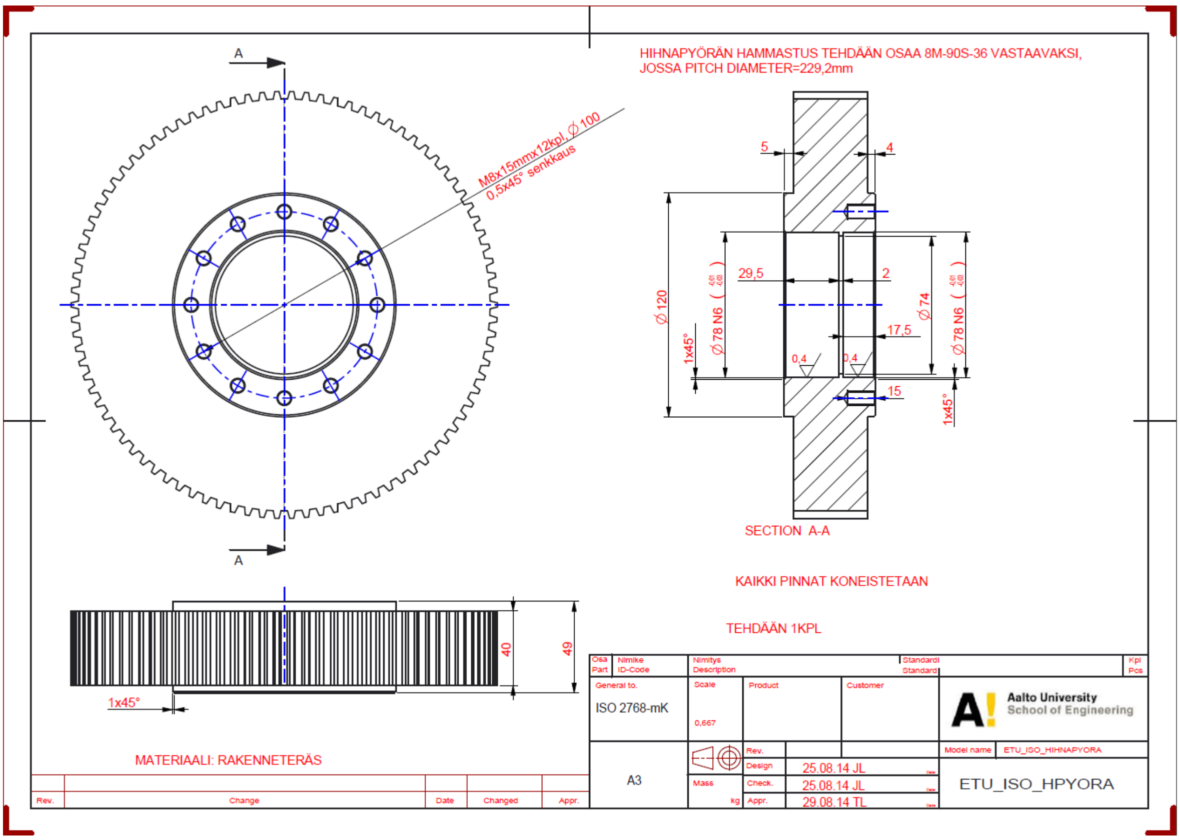


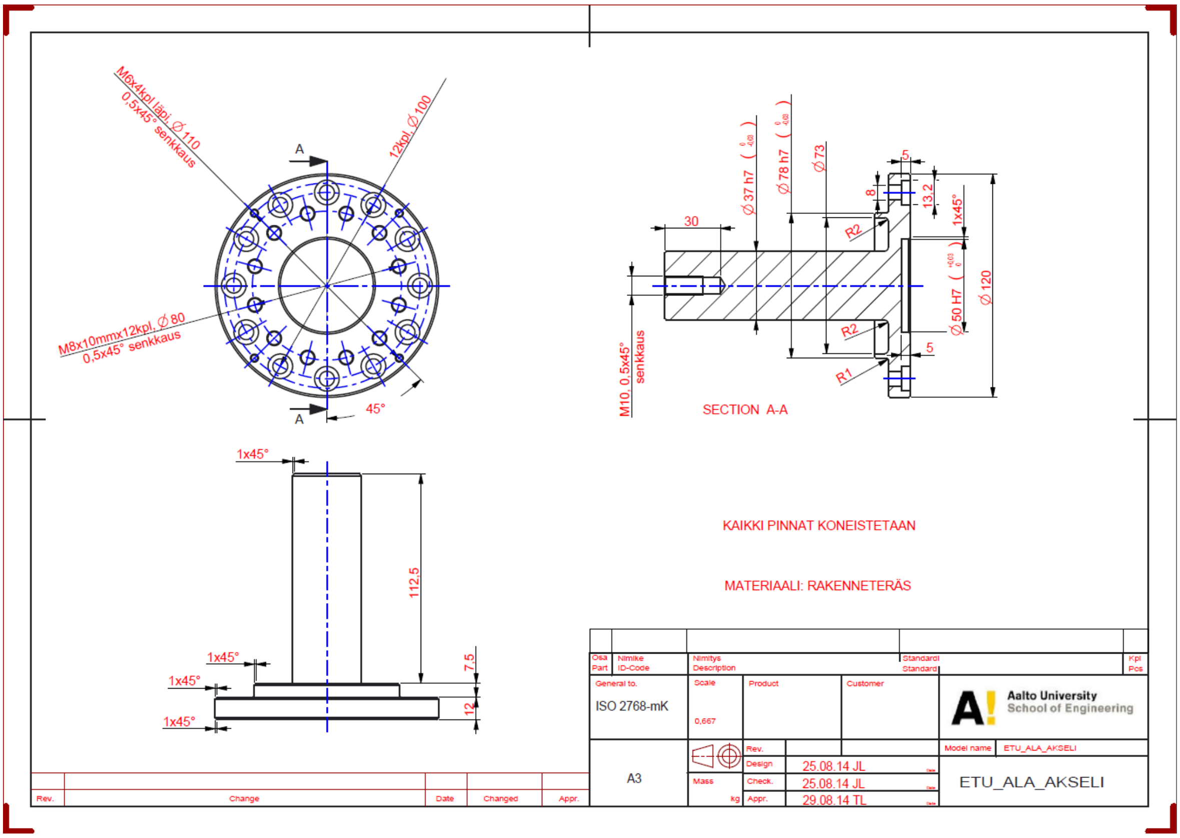
OSA LIITETÄÄN MOOTTORIIN: SIEMENS DRIVE MOTOR 1PV5135-4WS28 (67KW). SPLINEN TULEE OLLA YHTEENSOPIVA TÄMÄN MOOTTORIN AKSELIN SPLINEN KANSSA. MOOTTORIN DATASHEET LIITTEENÄ.
M8x12kpl läpi, Ø 80
0,5x45° senkkaus
M6x12kpl läpi, Ø 58
0,5x45° senkkaus
M4x4kpl läpi, Ø 46
0,5x45° senkkaus
SECTION A-A
KAIKKI PINNAT KONEISTETAAN
MATERIAALI: RAKENNETERÄS
ISO 2768-mK
A3
Aalto University
School of Engineering
SPLAINI_ETU_MOTILLE2

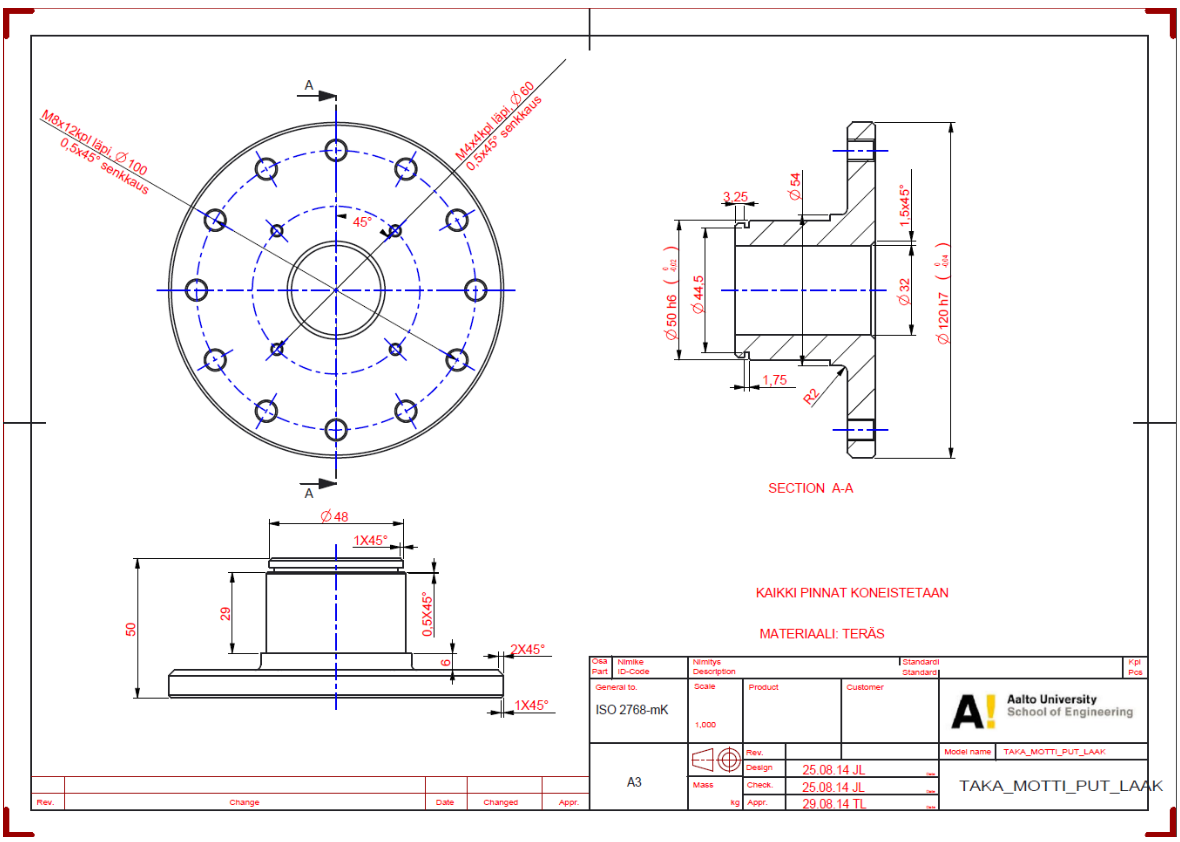

82,6
15
Ø 115
Ø 50
Ø 35
Ø 60
SECTION A-A
A
M8 x 12
molempiin laippoihin
Ø 80
General to.
ISO 2768-mK
Scale
1,000
Product
Customer
Aalto University
School of Engineering
Model name
VALIAKSELI_MUOKATTU
A3
Design 25.08.14 JL
Check. 25.08.14 JL
Appr. 29.08.14 TL
VALIAKSELI
Rev.
Change
Date
Changed
Appr.

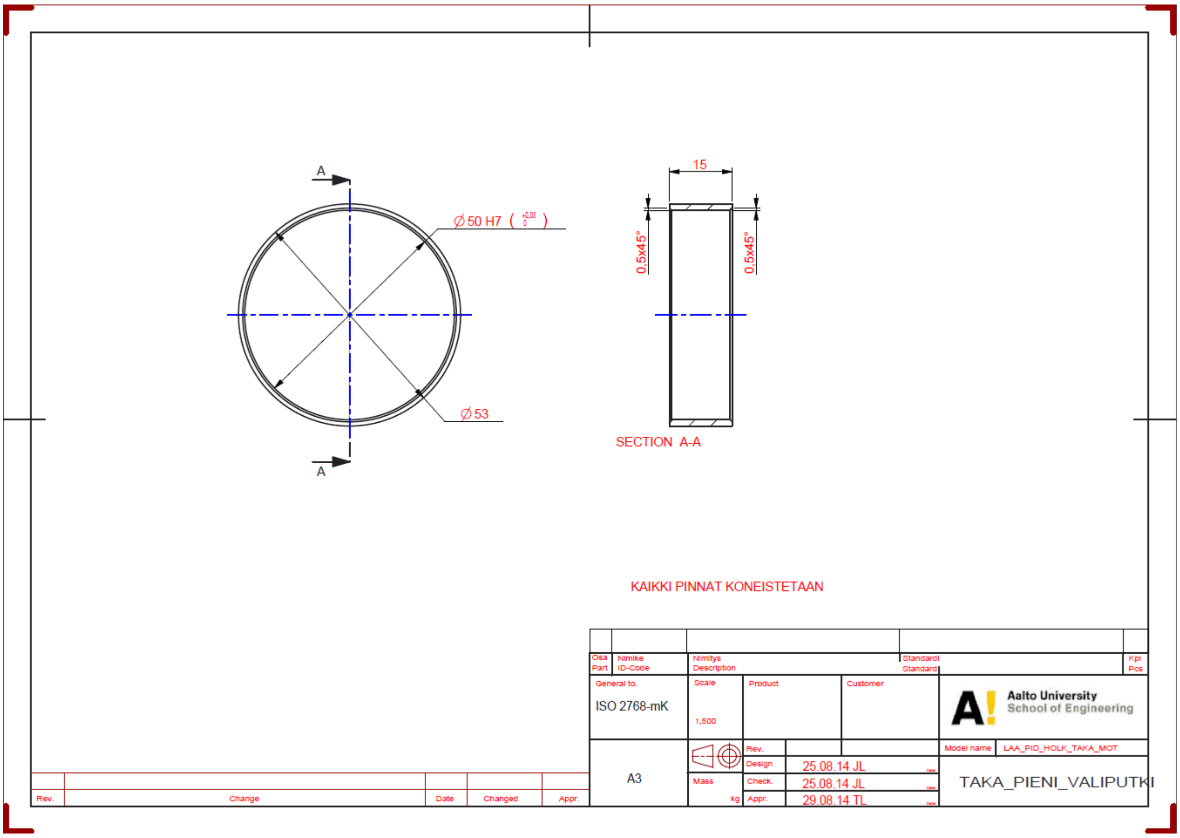

A
25
0.5x45°
0.5x45°
Ø 65 H7
SECTION A-A
General to.
ISO 2768-mK
Scale
0,667
Product
Customer
Aalto University
School of Engineering
Model name
KTR_ETU_HIHNAV_ALA
A3
Design 25.08.14 JL
Check. 25.08.14 JL
Appr. 29.08.14 TL
KTR_ETEEN
Rev.
Change
Date
Changed
Appr.

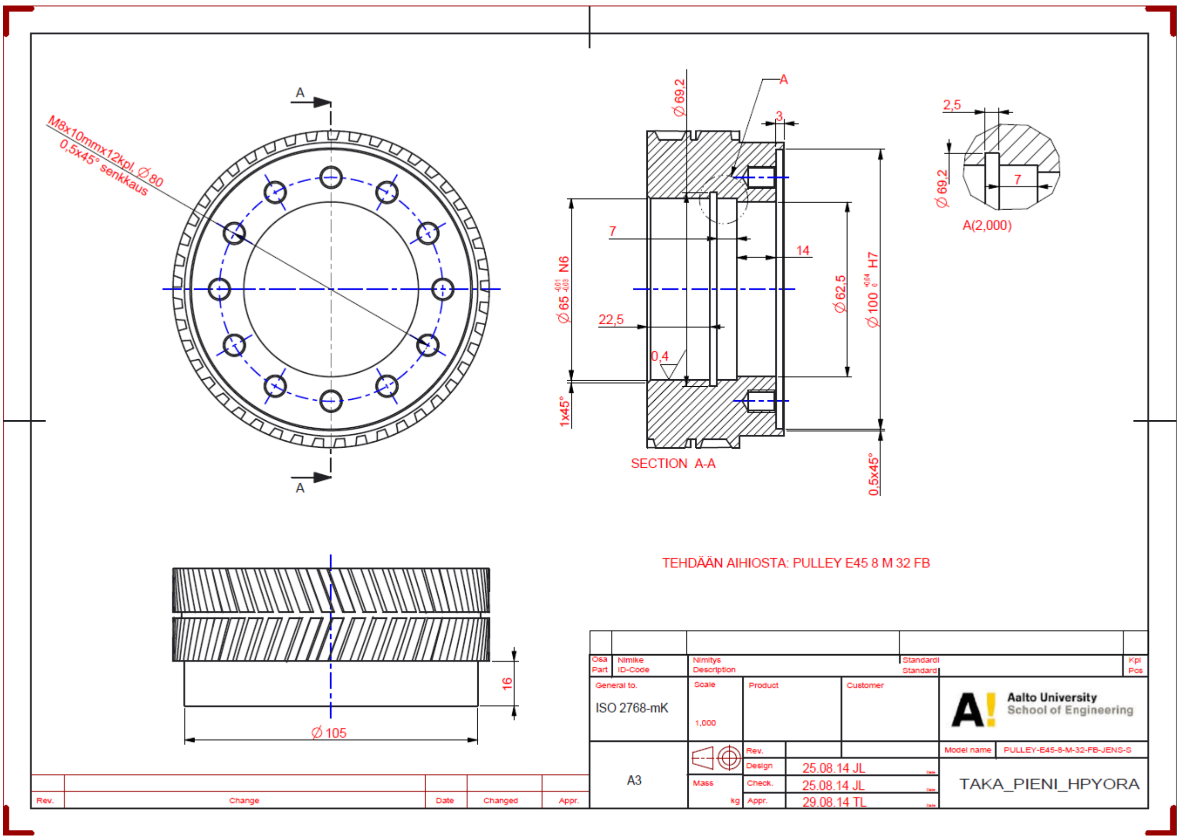

M4x4kpl läpi, ∅73
0,5x45° senkkaus
M8x12kpl läpi, ∅100
0,5x45° senkkaus
45°
SECTION A-A
KAIKKI PINNAT KONEISTETAAN
MATERIAALI: RAKENNETERÄS
ISO 2768-mK
0,667
A3
25.08.14 JL
25.08.14 JL
29.08.14 TL
Aalto University
School of Engineering
ETU_ALA_PUTKI_LAAKERILLE
ETU_ALA_LAAKERIPUTKI

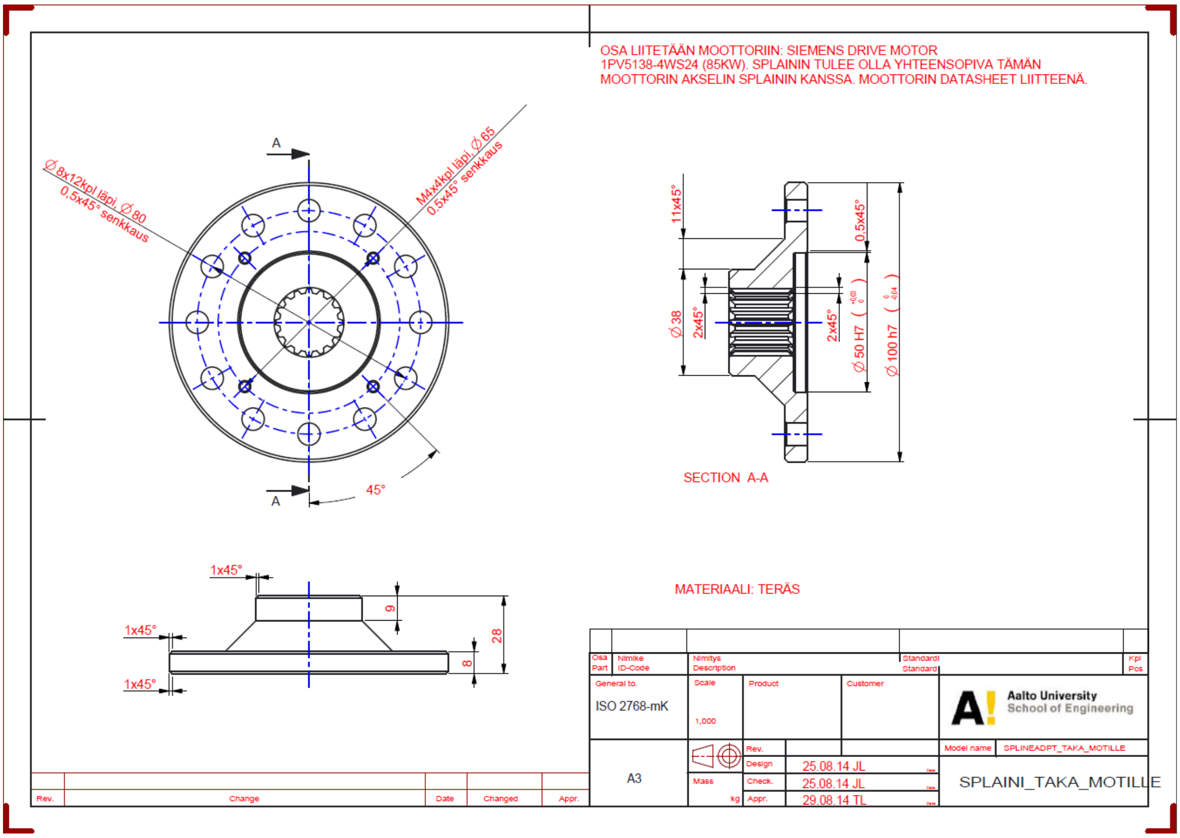

HIHNAPYÖRÄN HAMMASTUS TEHDÄÄN OSAA 8M-90S-36 VASTAAVAKSI,
JOSSA PITCH DIAMETER=229,2mm
M8x15mmx12kpl, ∅100
0,5x45° senkkaus
SECTION A-A
KAIKKI PINNAT KONEISTETAAN
TEHDÄÄN 1KPL
MATERIAALI: RAKENNETERÄS
ISO 2768-mK
0,667
A3
25.08.14 JL
25.08.14 JL
29.08.14 TL
Aalto University
School of Engineering
ETU_ISO_HIHNAPYORA
ETU_ISO_HPYORA

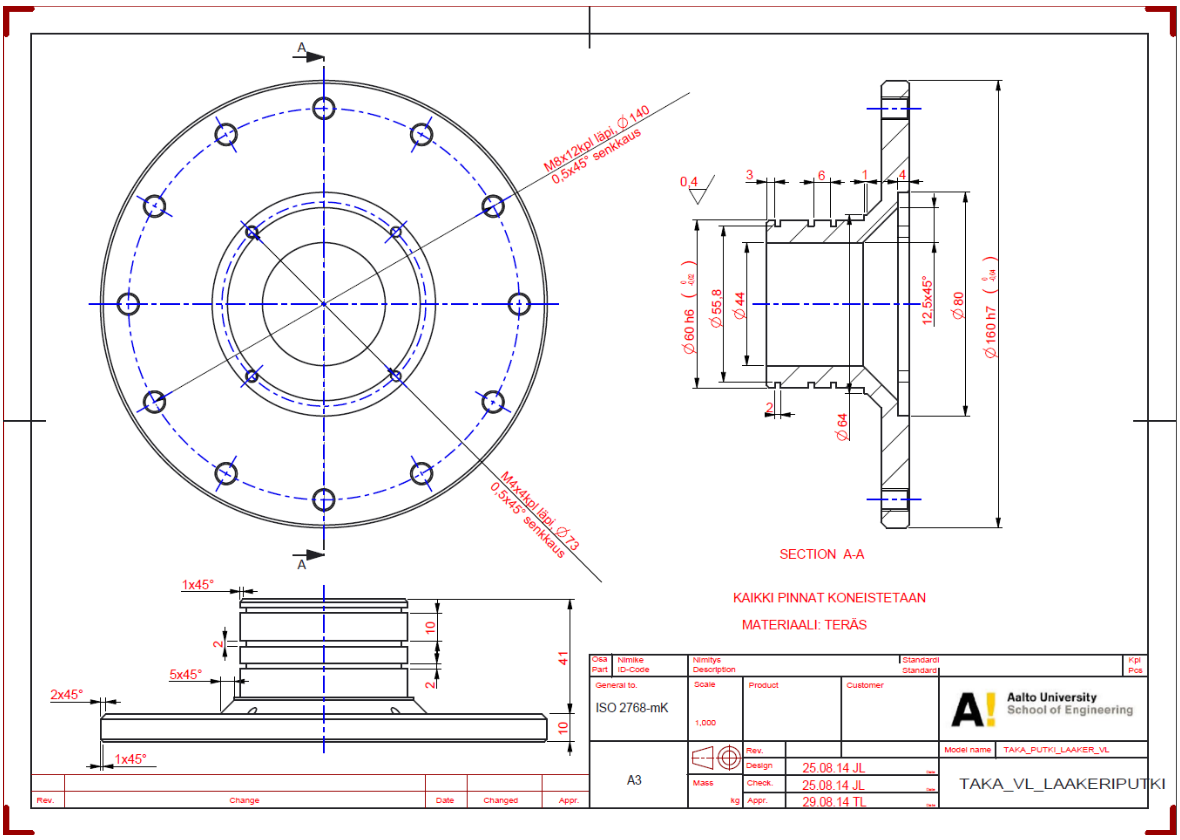

M6x4kpl läpi, ∅110
0,5x45° senkkaus
12kpl, ∅100
M8x10mmx12kpl, ∅80
0,5x45° senkkaus
45°
A
SECTION A-A
KAIKKI PINNAT KONEISTETAAN
MATERIAALI: RAKENNETERÄS
ISO 2768-mK
0,667
A3
25.08.14 JL
25.08.14 JL
29.08.14 TL
Aalto University
School of Engineering
ETU_ALA_AKSELI

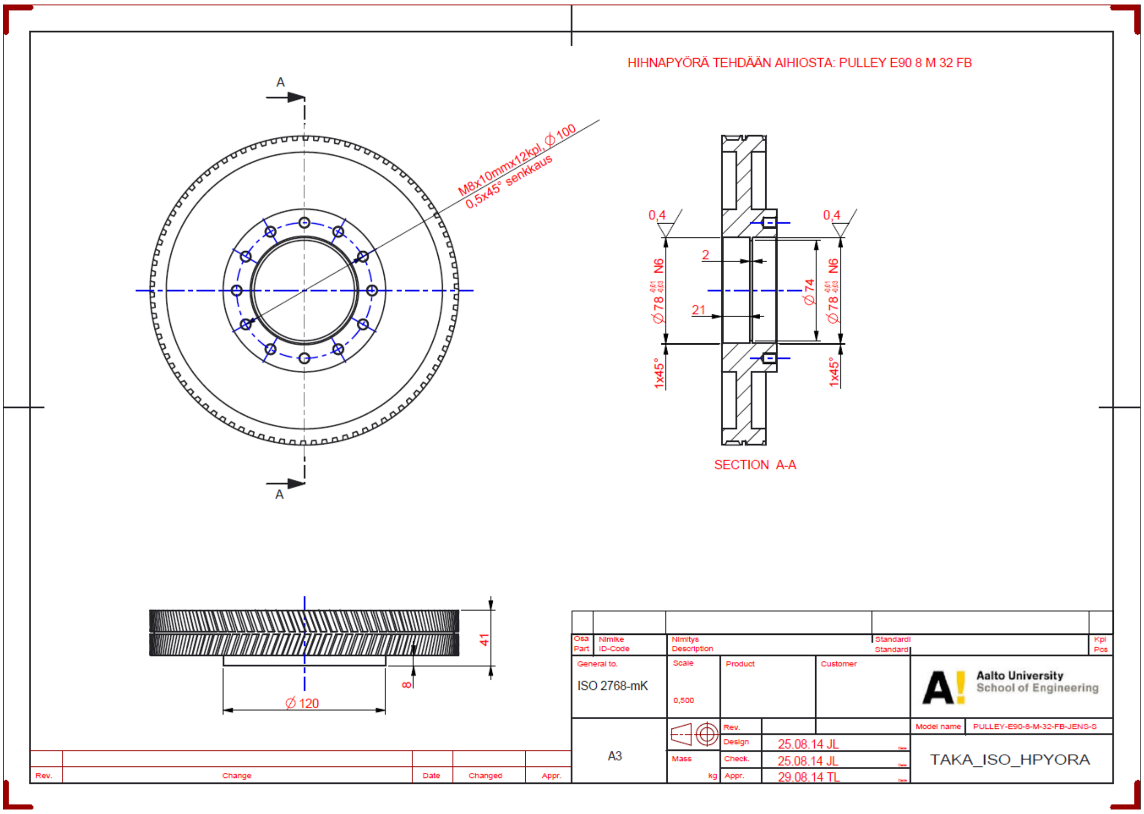

M8x12kpl läpi, ∅100
0,5x45° senkkaus
M4x4kpl läpi, ∅60
0,5x45° senkkaus
45°
A
SECTION A-A
KAIKKI PINNAT KONEISTETAAN
MATERIAALI: TERÄS
ISO 2768-mK
1,000
A3
25.08.14 JL
25.08.14 JL
29.08.14 TL
Aalto University
School of Engineering
TAKA_MOTTI_PUT_LAAK

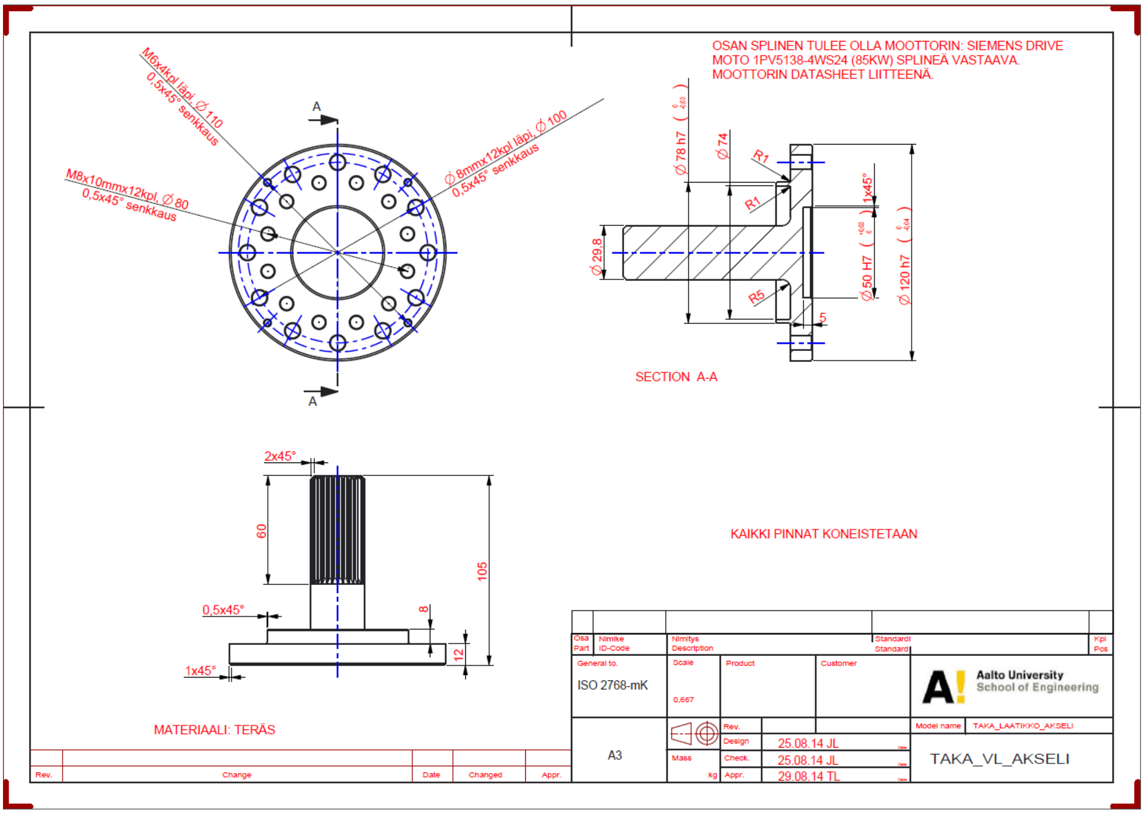

A
∅ 50 H7
∅ 53
15
0,5x45°
0,5x45°
SECTION A-A
KAIKKI PINNAT KONEISTETAAN
ISO 2768-mK
A3
Aalto University
School of Engineering
LAA_PID_HOLK_TAKA_MOT
25.08.14 JL
25.08.14 JL
29.08.14 TL
TAKA_PIENI_VALIPUTKI

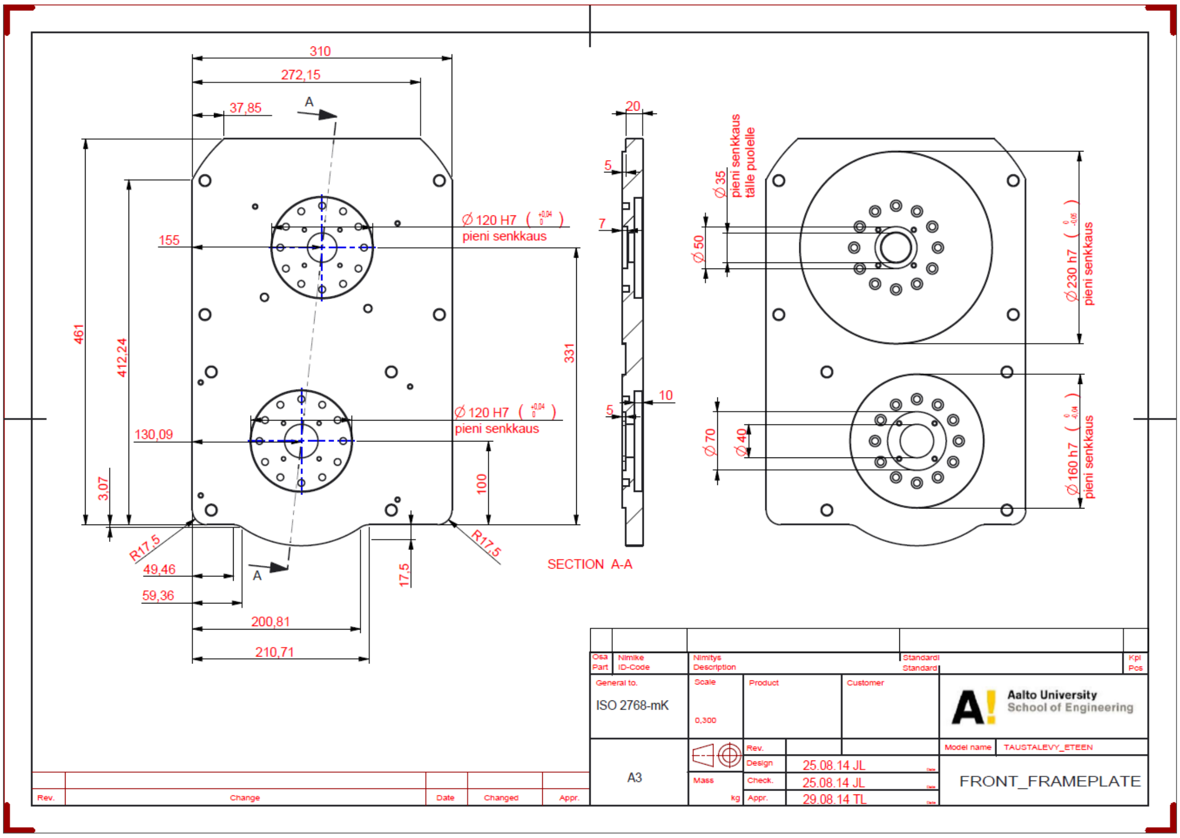

A
M8x10mmx12kpl ∅ 80
0,5x45° senkkaus
∅ 69,2
3
7
14
22,5
0,4
∅ 65 N6
∅ 62,5
∅ 100 H7
1x45°
0,5x45°
SECTION A-A
2,5
∅ 69,2
7
A(2,000)
TEHDÄÄN AIHIOSTA: PULLEY E45 8 M 32 FB
16
∅ 105
ISO 2768-mK
A3
Aalto University
School of Engineering
25.08.14 JL
25.08.14 JL
29.08.14 TL
TAKA_PIENI_HPYORA

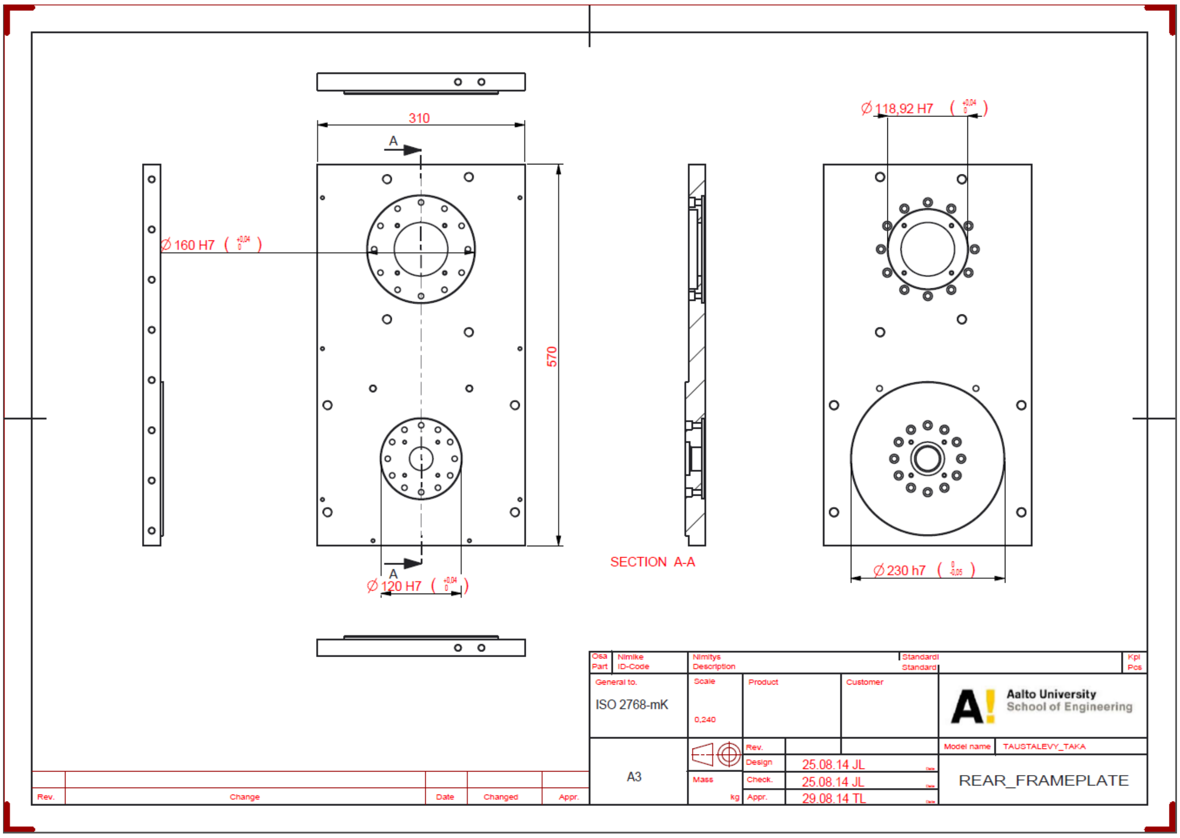

OSA LIITETÄÄN MOOTTORIIN: SIEMENS DRIVE MOTOR
1PV5138-4WS24 (85KW). SPLAININ TULEE OLLA YHTEENSOPIVA TÄMÄN
MOOTTORIN AKSELIN SPLAININ KANSSA. MOOTTORIN DATASHEET LIITTEENÄ.
⌀8x12kpl läpi, ⌀80
0,5x45° senkkaus
M4x4kpl läpi, ⌀65
0,5x45° senkkaus
45°
SECTION A-A
MATERIAALI: TERÄS
ISO 2768-mK
A3
25.08.14 JL
25.08.14 JL
29.08.14 TL
SPLAINI_TAKA_MOTILLE
Aalto University
School of Engineering

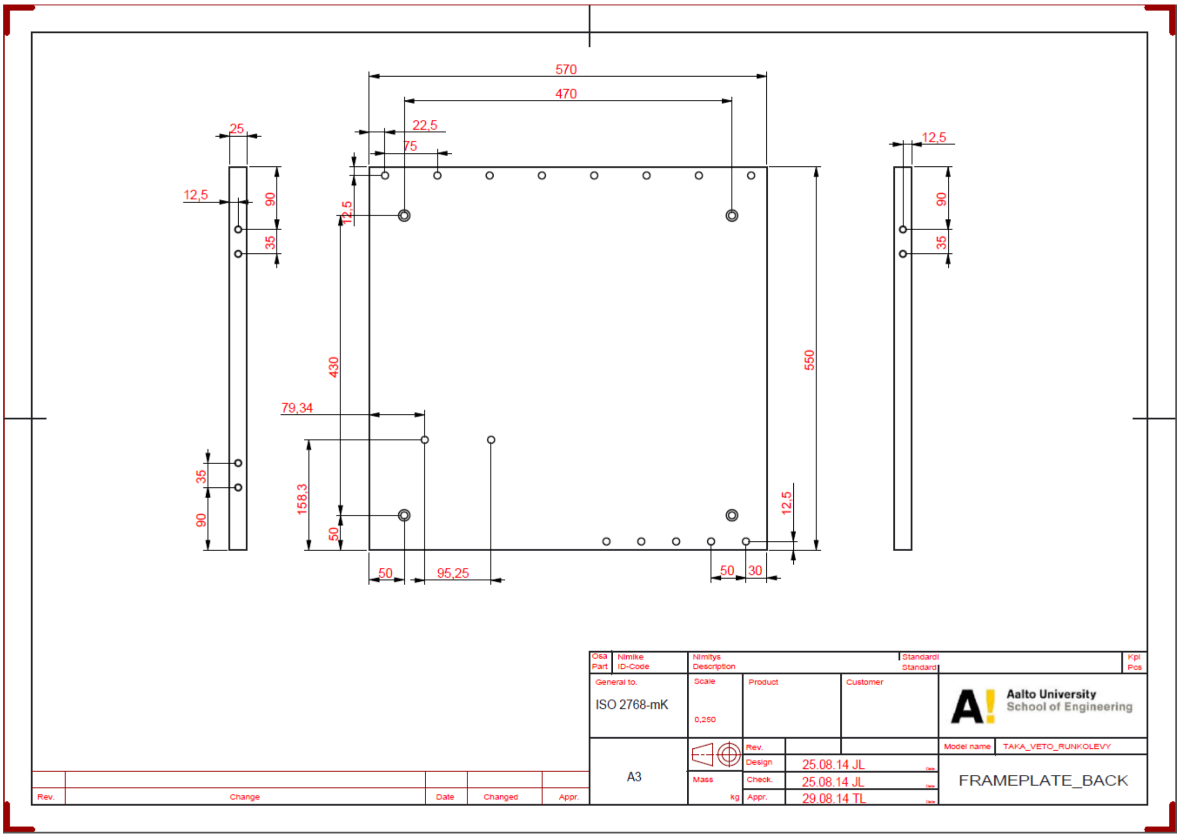

M8x12kpl läpi, ⌀140
0,5x45° senkkaus
M4x4kpl läpi, ⌀73
0,5x45° senkkaus
SECTION A-A
KAIKKI PINNAT KONEISTETAAN
MATERIAALI: TERÄS
ISO 2768-mK
A3
25.08.14 JL
25.08.14 JL
29.08.14 TL
TAKA_VL_LAAKERIPUTKI
Aalto University
School of Engineering

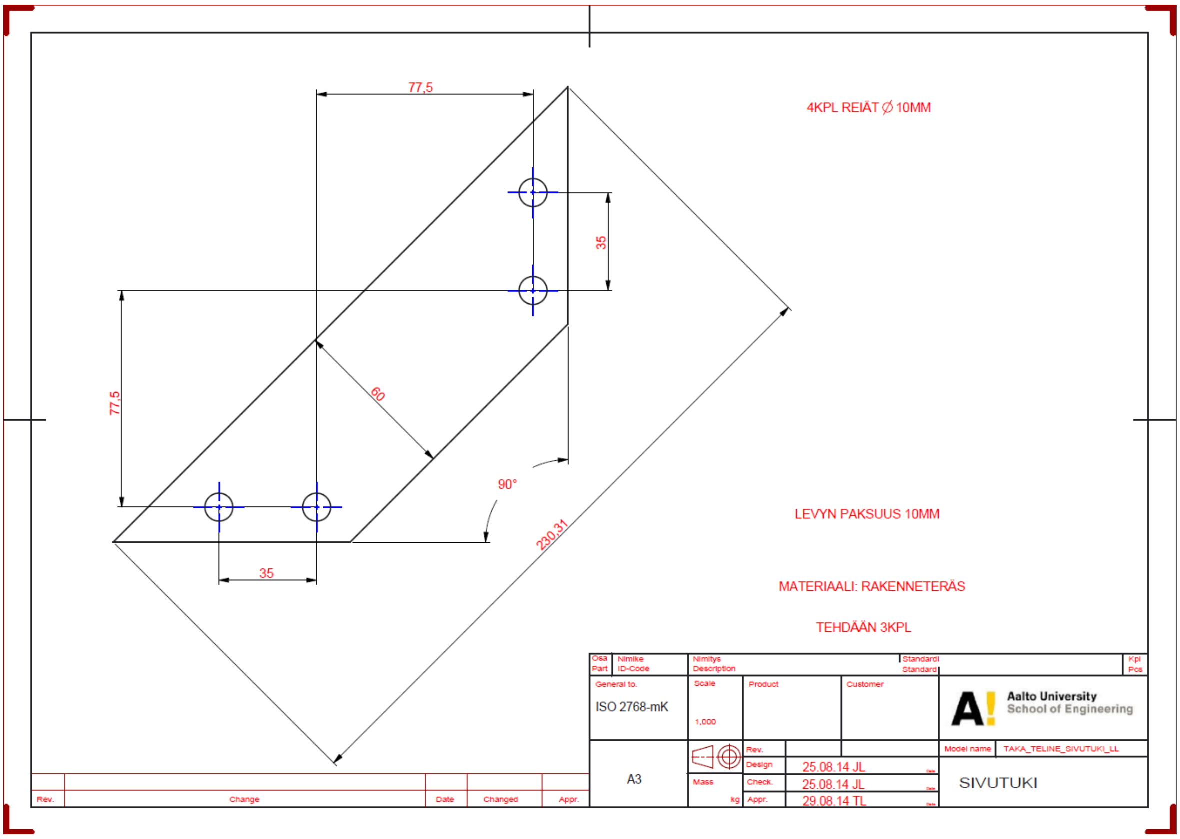

HIHNAPYÖRÄ TEHDÄÄN AIHIOSTA: PULLEY E90 8 M 32 FB
M8x10mmx12kpl, Ø 100
0,5x45° senkkaus
SECTION A-A
Ø 120
41
8
General to.
ISO 2768-mK
Scale
0,500
A3
Design 25.08.14 JL
Check. 25.08.14 JL
Appr. 29.08.14 TL
Aalto University
School of Engineering
Model name PULLEY-E90-8-M-32-FB-JENS-S
TAKA_ISO_HPYORA

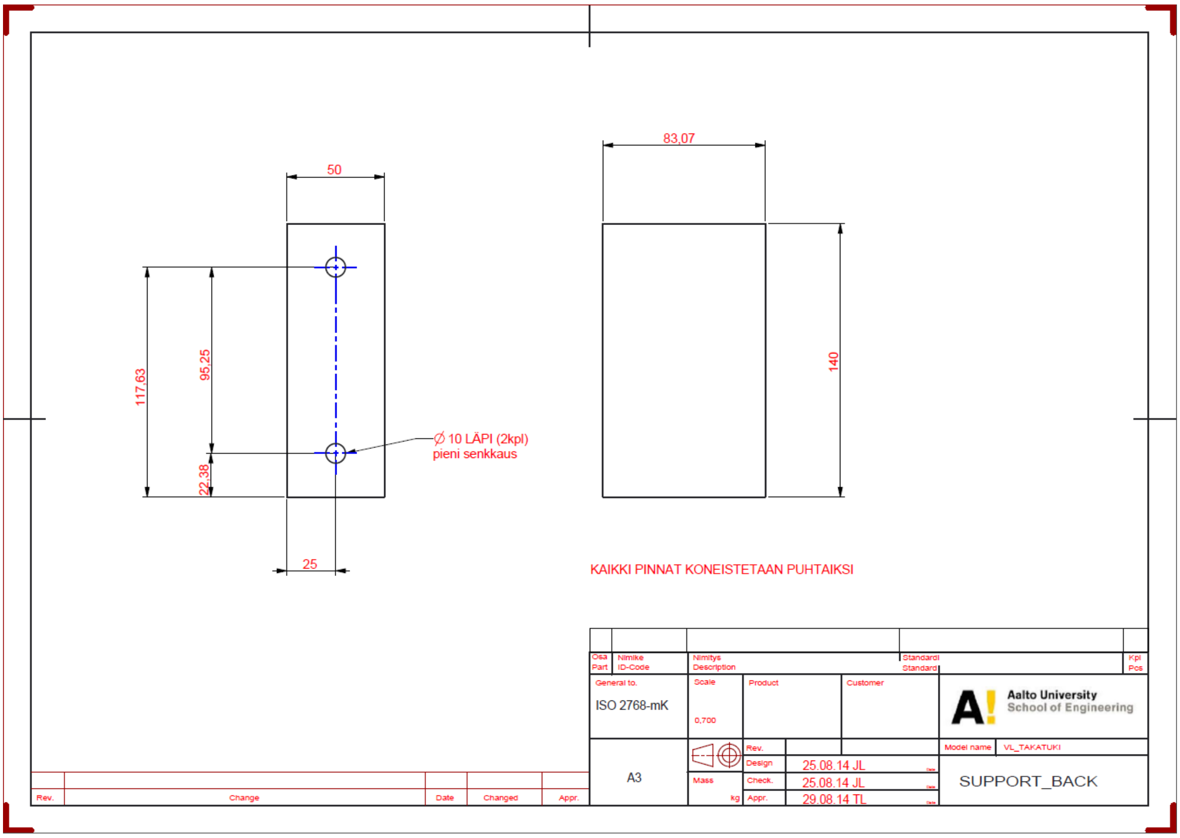

OSAN SPLINEN TULEE OLLA MOOTTORIN: SIEMENS DRIVE
MOTO 1PV5138-4WS24 (85KW) SPLINEÄ VASTAAVA.
MOOTTORIN DATASHEET LIITTEENÄ.
M6x4kpl läpi, Ø 110
0,5x45° senkkaus
M8x10mmx12kpl, Ø 80
0,5x45° senkkaus
Ø 8mmx12kpl läpi, Ø 100
0,5x45° senkkaus
Ø 78 h7
Ø 74
Ø 29,8
Ø 50 H7
Ø 120 h7
R1
R5
1x45°
5
SECTION A-A
KAIKKI PINNAT KONEISTETAAN
2x45°
60
105
0,5x45°
8
12
1x45°
MATERIAALI: TERÄS
General to.
ISO 2768-mK
Scale
0,667
A3
Design 25.08.14 JL
Check. 25.08.14 JL
Appr. 29.08.14 TL
Aalto University
School of Engineering
Model name TAKA_LAATIKKO_AKSELI
TAKA_VL_AKSELI

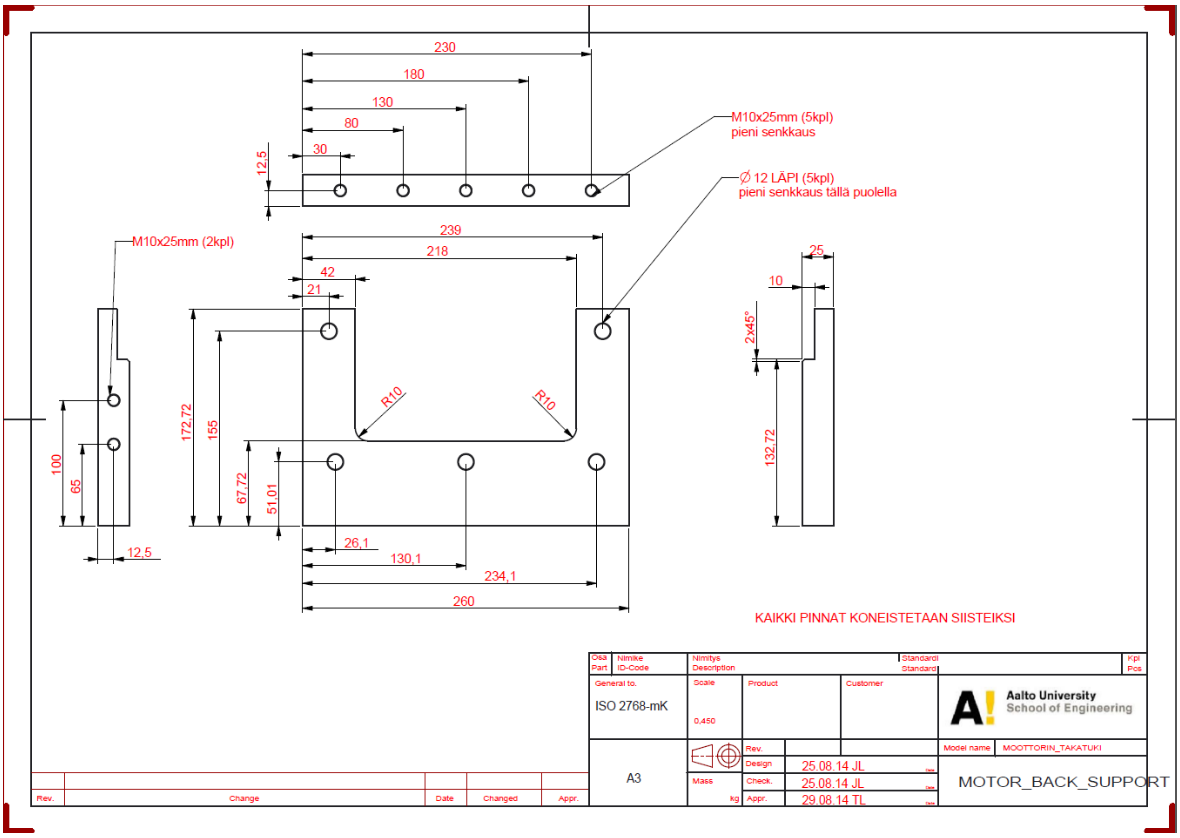

310
272,15
37,85
461
412,24
155
130,09
3,07
331
100
17,5
R17,5
49,46
59,36
200,81
210,71
Ø 120 H7
pieni senkkaus
20
5
7
10
SECTION A-A
Ø 35
pieni senkkaus
tälle puolelle
Ø 50
Ø 70
Ø 40
Ø 230 h7
Ø 160 h7
ISO 2768-mK
0,300
A3
25.08.14 JL
25.08.14 JL
29.08.14 TL
Aalto University
School of Engineering
TAUSTALEVY_ETEEN
FRONT_FRAMEPLATE

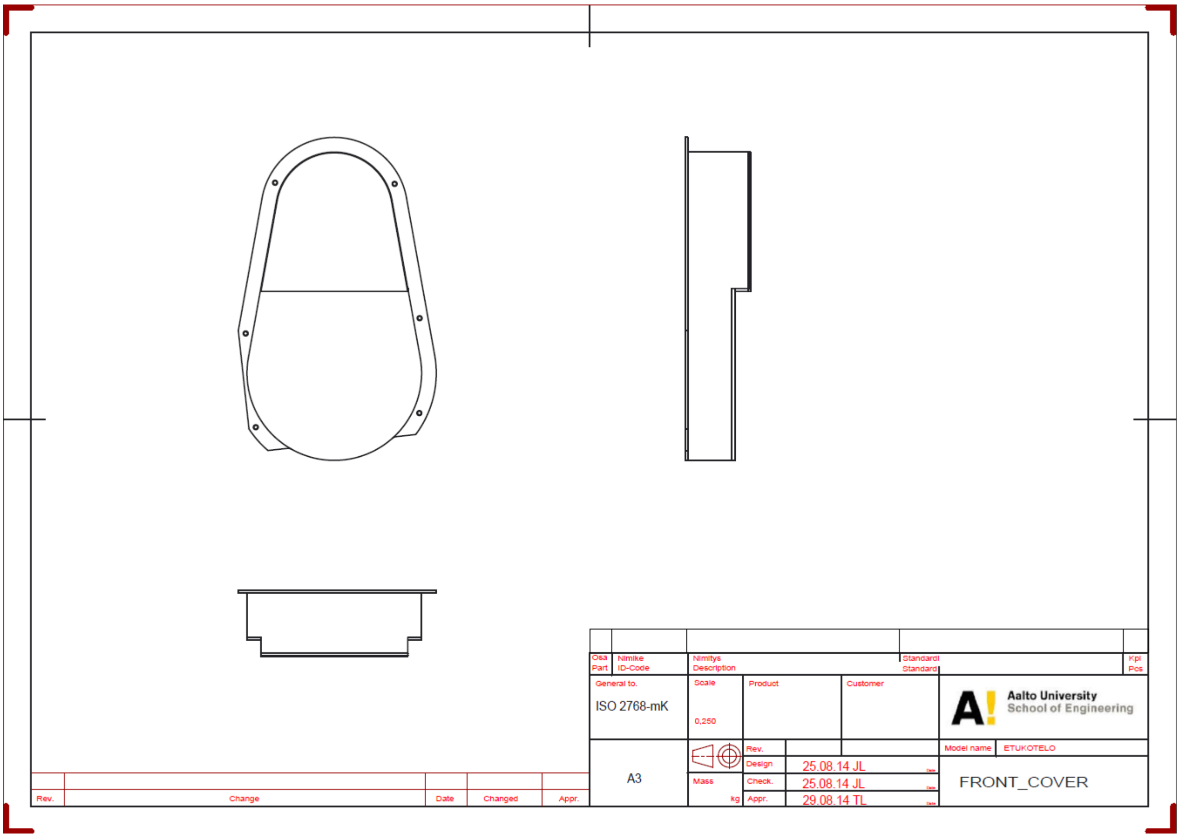

310
570
Ø 160 H7
Ø 120 H7
Ø 118,92 H7
Ø 230 h7
SECTION A-A
ISO 2768-mK
0,240
A3
25.08.14 JL
25.08.14 JL
29.08.14 TL
Aalto University
School of Engineering
TAUSTALEVY_TAKA
REAR_FRAMEPLATE

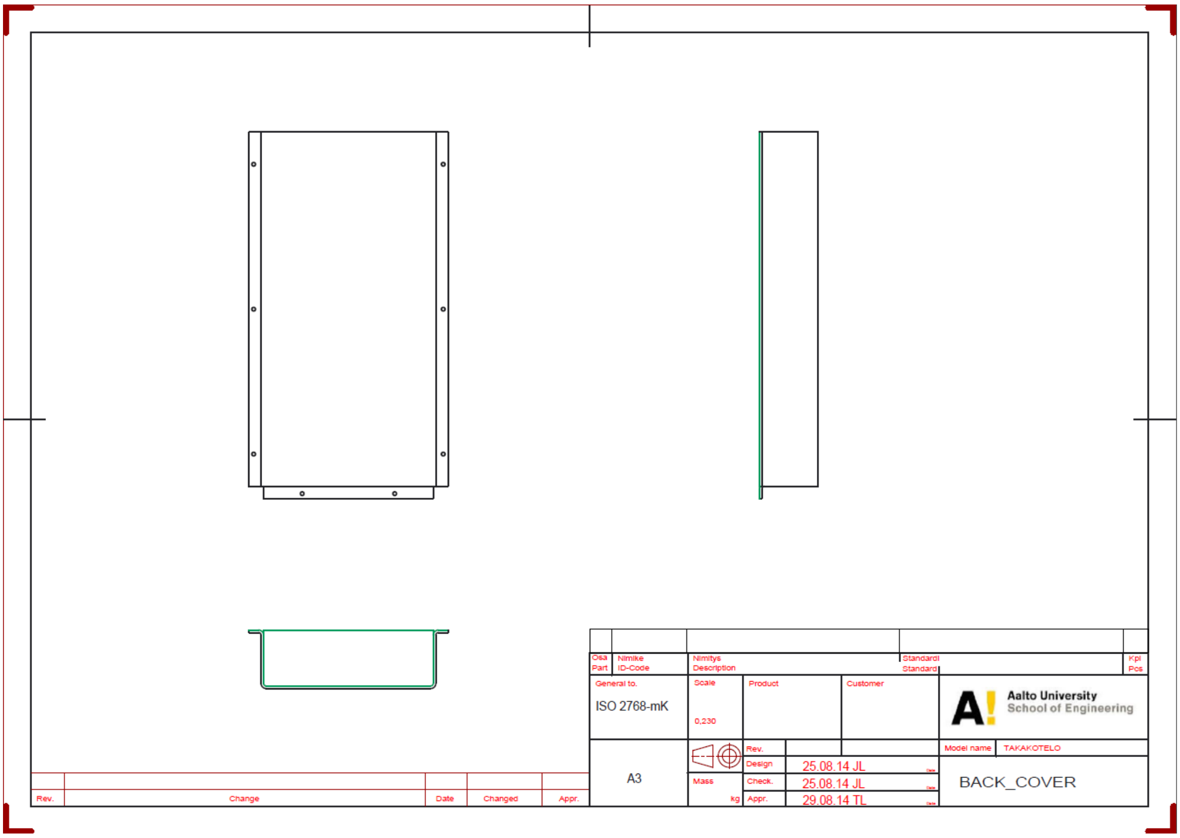

570
470
22,5
75
25
12,5
90
35
12,5
430
79,34
158,3
50
550
12,5
50
95,25
50
30
12,5
90
35
35
90
Osa Part
Nimike ID-Code
Nimitys Description
Standard Standard
Kpl Pos
General to.
ISO 2768-mK
Scale
0,250
Product
Customer
Aalto University
School of Engineering
A3
Rev.
Design
Check.
Appr.
Mass
kg
25.08.14 JL
25.08.14 JL
29.08.14 TL
Model name
TAKA_VETO_RUNKOLEVY
FRAMEPLATE_BACK
Rev.
Change
Date
Changed
Appr.

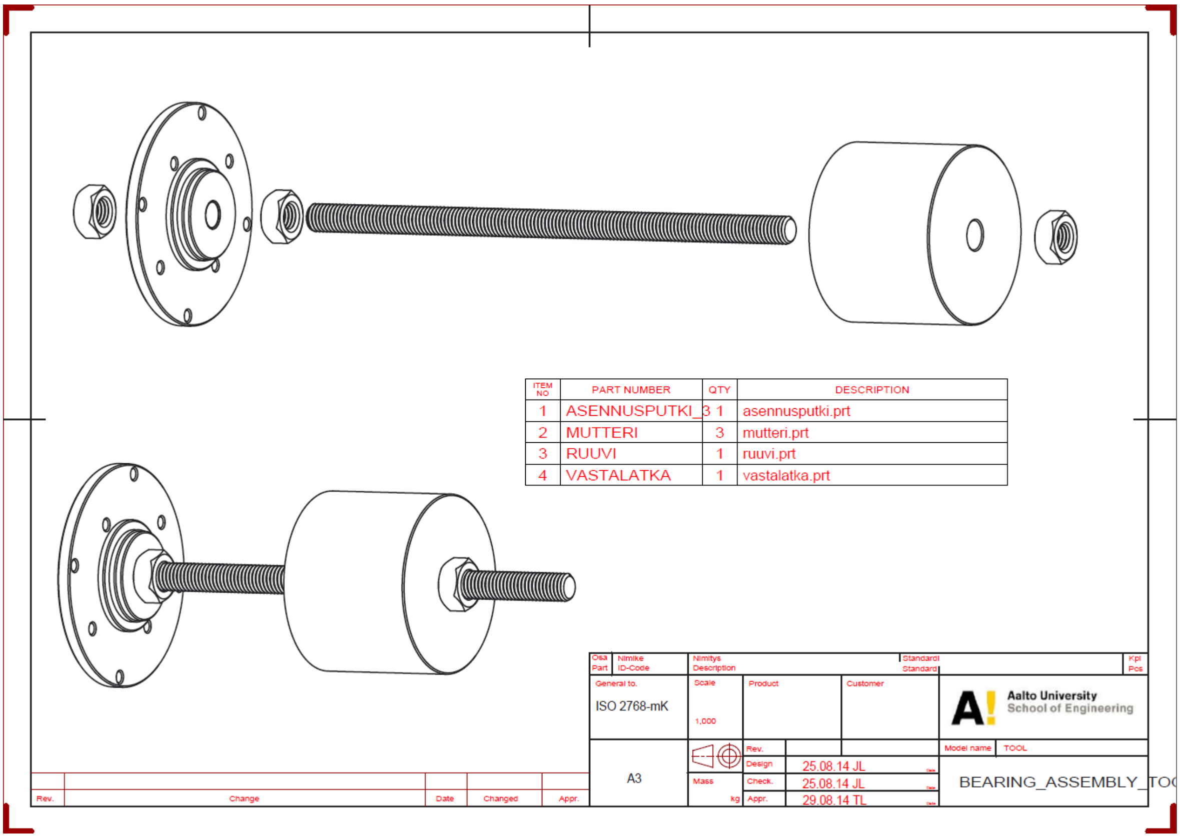

77,5
4KPL REIÄT ∅ 10MM
35
77,5
60
90°
230,31
35
LEVYN PAKSUUS 10MM
MATERIAALI: RAKENNETERÄS
TEHDÄÄN 3KPL
Osa Part
Nimike ID-Code
Nimitys Description
Standard Standard
Kpl Pos
General to.
ISO 2768-mK
Scale
1,000
Product
Customer
Aalto University
School of Engineering
A3
Rev.
Design
Check.
Appr.
Mass
kg
25.08.14 JL
25.08.14 JL
29.08.14 TL
Model name
TAKA_TELINE_SIVUTUKI_LL
SIVUTUKI
Rev.
Change
Date
Changed
Appr.

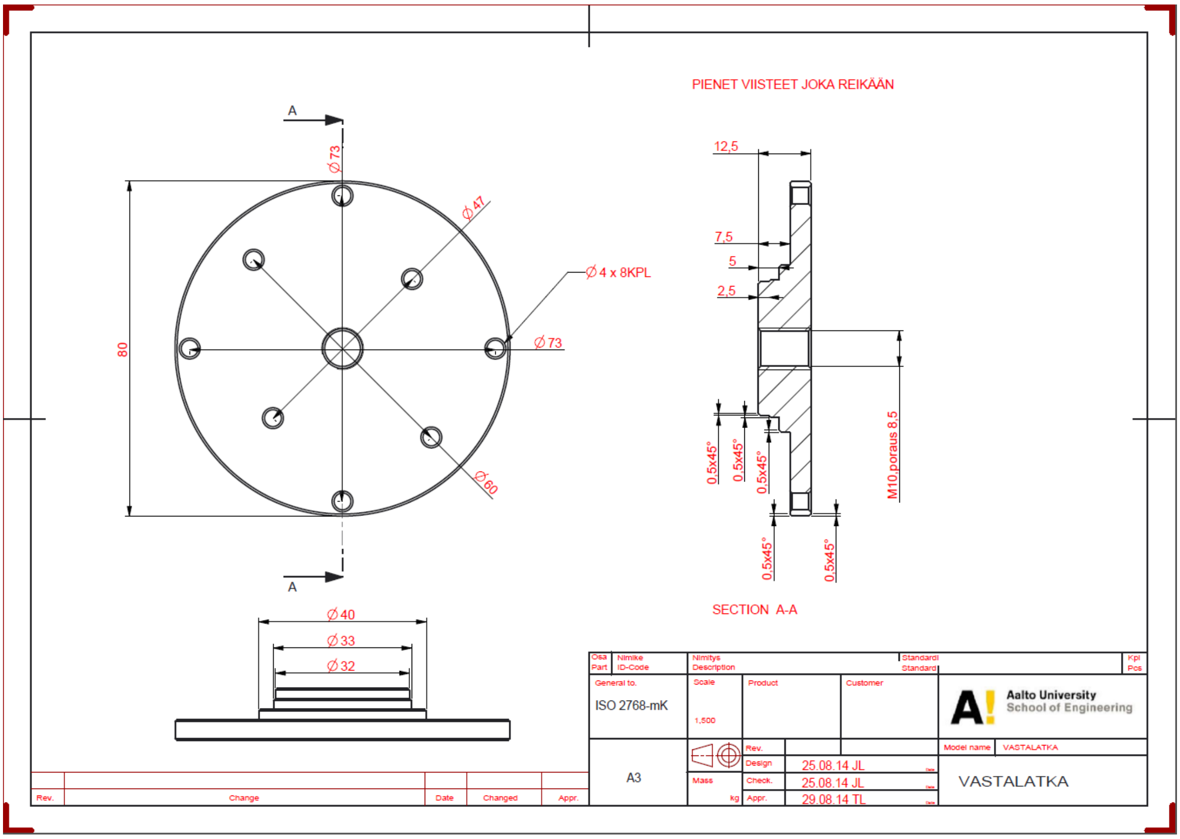

50
83,07
117,63
95,25
22,38
140
25
∅ 10 LÄPI (2kpl)
pieni senkkaus
KAIKKI PINNAT KONEISTETAAN PUHTAIKSI
Osa Part
Nimike ID-Code
Nimitys Description
Standardi Standard
Kpl Pcs
General to.
ISO 2768-mK
Scale
0,700
Product
Customer
Aalto University
School of Engineering
A3
Mass
kg
Rev.
Design
Check.
Appr.
25.08.14 JL
25.08.14 JL
29.08.14 TL
Model name
VL_TAKATUKI
SUPPORT_BACK
Rev.
Change
Date
Changed
Appr.

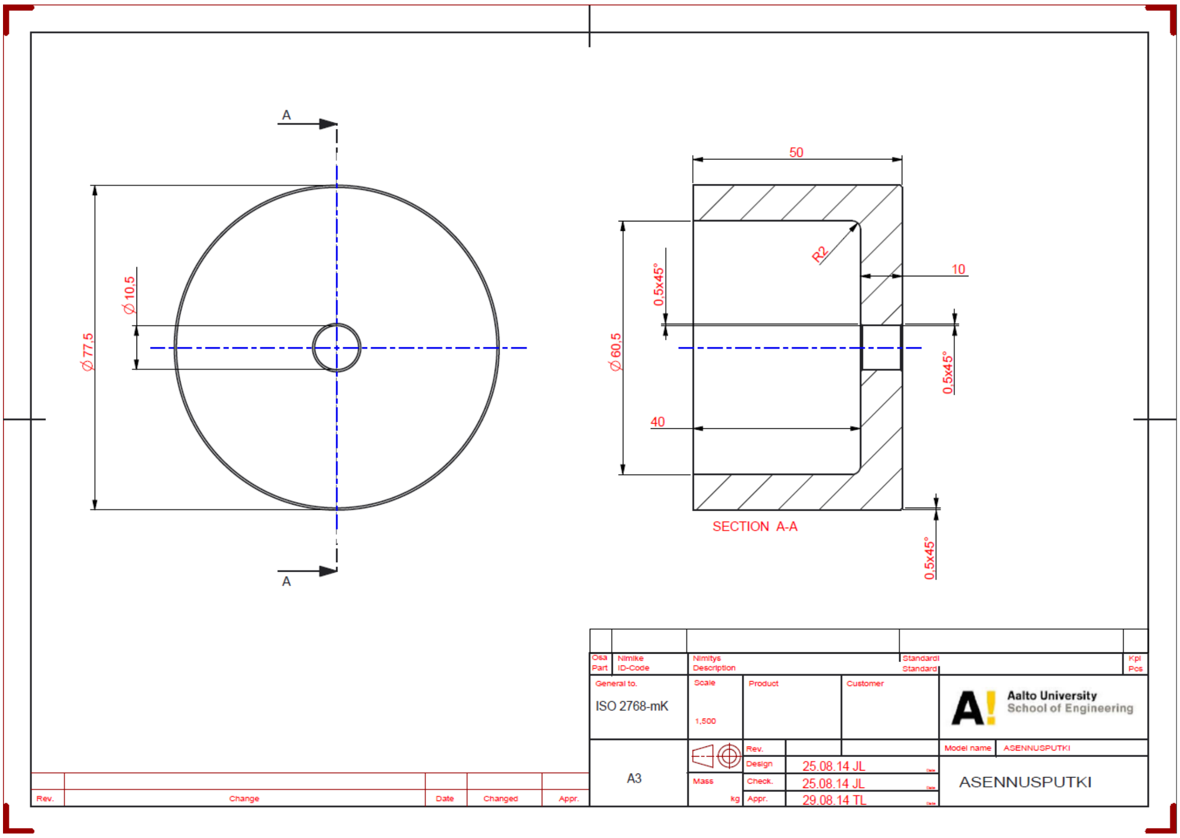

230
180
130
80
30
12,5
M10x25mm (5kpl)
pieni senkkaus
∅ 12 LÄPI (5kpl)
pieni senkkaus tällä puolella
M10x25mm (2kpl)
239
218
42
21
25
10
2x45°
R10
R10
172,72
155
67,72
51,01
100
65
132,72
12,5
26,1
130,1
234,1
260
KAIKKI PINNAT KONEISTETAAN SIISTEIKSI
Osa Part
Nimike ID-Code
Nimitys Description
Standardi Standard
Kpl Pcs
General to.
ISO 2768-mK
Scale
0,450
Product
Customer
Aalto University
School of Engineering
A3
Mass
kg
Rev.
Design
Check.
Appr.
25.08.14 JL
25.08.14 JL
29.08.14 TL
Model name
MOOTTORIN_TAKATUKI
MOTOR_BACK_SUPPORT
Rev.
Change
Date
Changed
Appr.

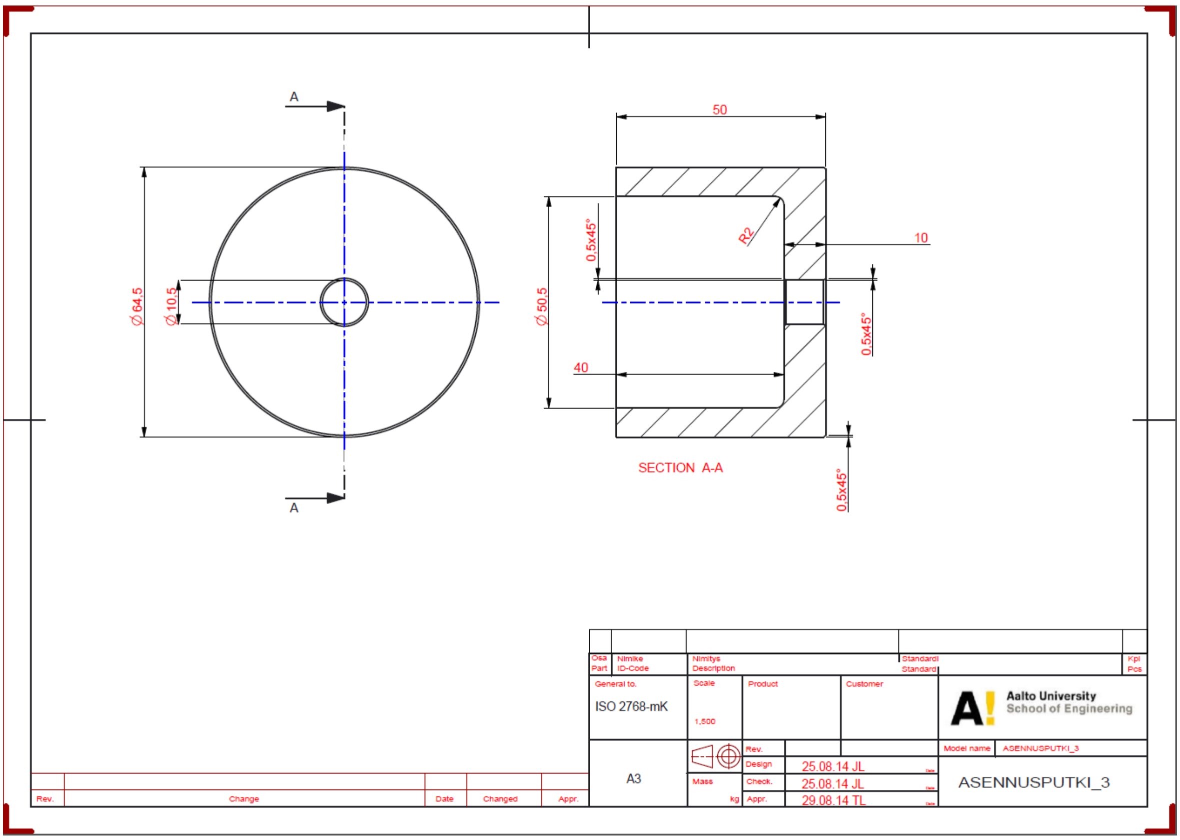

Osa Part
Nimike ID-Code
Nimitys Description
Standardi Standard
Kpl Pos
General to.
ISO 2768-mK
Scale
0,250
Product
Customer
Aalto University
School of Engineering
A3
Rev.
Design
Check.
Appr.
Mass
kg
25.08.14 JL
25.08.14 JL
29.08.14 TL
Model name
ETUKOTELO
FRONT_COVER
Rev.
Change
Date
Changed
Appr.

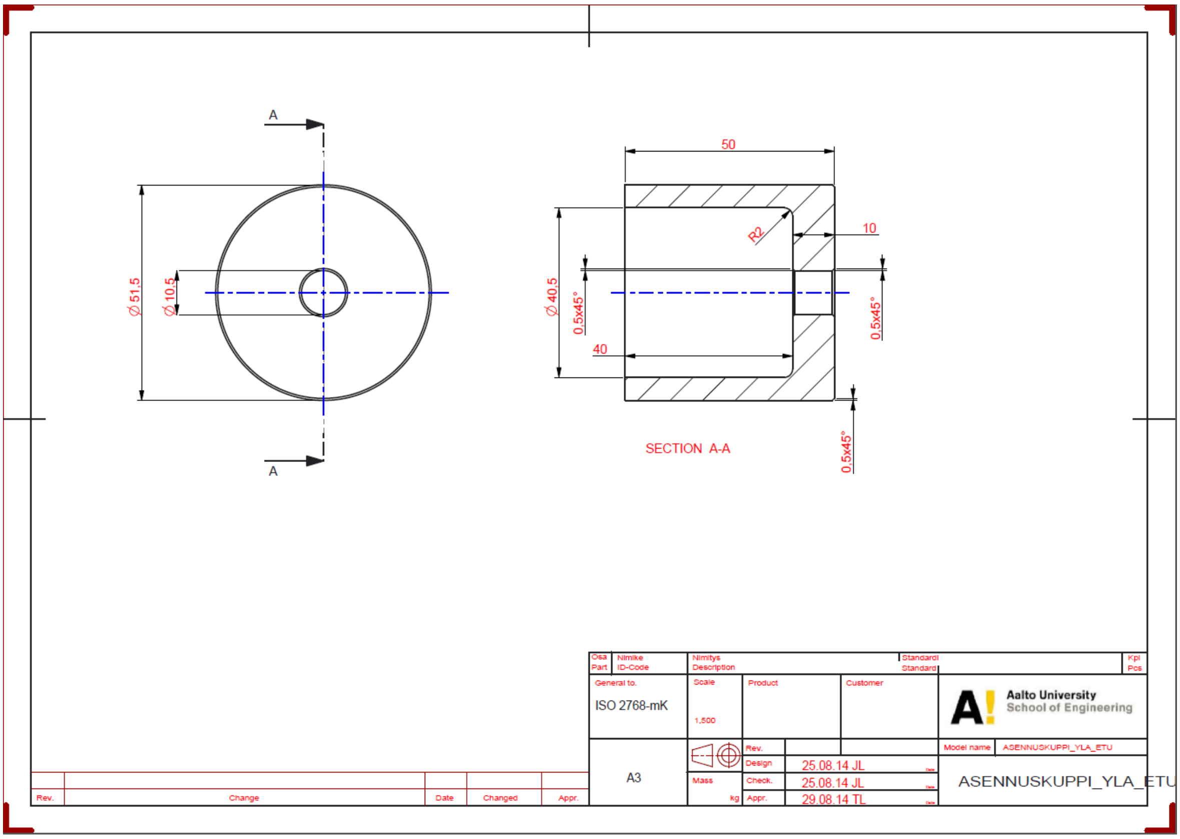

Osa Part
Nimike ID-Code
Nimitys Description
Standardi Standard
Kpl Pos
General to.
ISO 2768-mK
Scale
0,230
Product
Customer
Aalto University
School of Engineering
A3
Rev.
Design
Check.
Appr.
Mass
kg
25.08.14 JL
25.08.14 JL
29.08.14 TL
Model name
TAKAKOTELO
BACK_COVER
Rev.
Change
Date
Changed
Appr.


ITEM NO
PART NUMBER
QTY
DESCRIPTION
1 ASENNUSPUTKI_3 1 asennusputki.prt
2 MUTTERI 3 mutteri.prt
3 RUUVI 1 ruuvi.prt
4 VASTALATKA 1 vastalatka.prt
Osa Part
Nimike ID-Code
Nimitys Description
Standardi Standard
Kpl Pcs
General to.
ISO 2768-mK
Scale
1,000
Product
Customer
Aalto University
School of Engineering
Model name
TOOL
Rev.
Design
25.08.14 JL
Check.
25.08.14 JL
Appr.
29.08.14 TL
A3
Mass
kg
BEARING_ASSEMBLY_TOO
Rev.
Change
Date
Changed
Appr.

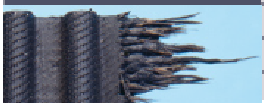
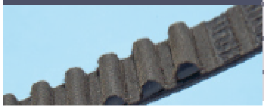
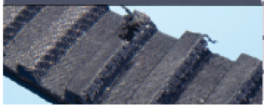
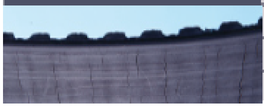
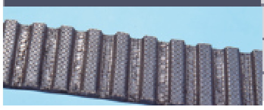

PIENET VIISTEET JOKA REIKÄÄN
A
Ø 73
Ø 47
Ø 4 x 8KPL
Ø 73
Ø 60
80
A
Ø 40
Ø 33
Ø 32
12,5
7,5
5
2,5
0,5x45°
0,5x45°
0,5x45°
0,5x45°
0,5x45°
M10,poraus 8,5
SECTION A-A
Osa Part
Nimike ID-Code
Nimitys Description
Standardi Standard
Kpl Pcs
General to.
ISO 2768-mK
Scale
1,500
Product
Customer
Aalto University
School of Engineering
Model name
VASTALATKA
Rev.
Design
25.08.14 JL
Check.
25.08.14 JL
Appr.
29.08.14 TL
A3
Mass
kg
VASTALATKA
Rev.
Change
Date
Changed
Appr.


A
A
Ø77,5
Ø10,5
50
R2
10
0,5x45°
Ø60,5
40
SECTION A-A
0,5x45°
0,5x45°
Osa Part
Nimike ID-Code
Nimitys Description
Standardi Standard
Kpl Pos
General to.
ISO 2768-mK
Scale
1,500
Product
Customer
Aalto University
School of Engineering
A3
Rev.
Design
25.08.14 JL
Check.
25.08.14 JL
Appr.
29.08.14 TL
Mass
kg
Model name
ASENNUSPUTKI
ASENNUSPUTKI
Rev.
Change
Date
Changed
Appr.


A
A
Ø64,5
Ø10,5
50
R2
10
0,5x45°
Ø50,5
40
SECTION A-A
0,5x45°
0,5x45°
Osa Part
Nimike ID-Code
Nimitys Description
Standardi Standard
Kpl Pos
General to.
ISO 2768-mK
Scale
1,500
Product
Customer
Aalto University
School of Engineering
A3
Rev.
Design
25.08.14 JL
Check.
25.08.14 JL
Appr.
29.08.14 TL
Mass
kg
Model name
ASENNUSPUTKI_3
ASENNUSPUTKI_3
Rev.
Change
Date
Changed
Appr.


A
50
R2
10
⌀ 51.5
⌀ 10.5
⌀ 40.5
0.5x45°
40
0.5x45°
A
SECTION A-A
0.5x45°
General to.
ISO 2768-mK
Scale
1:500
Product
Customer
Aalto University
School of Engineering
A3
Mass
kg
Rev.
Design
25.08.14 JL
Check.
25.08.14 JL
Appr.
29.08.14 TL
Model name
ASENNUSKUPPI_YLA_ETU
ASENNUSKUPPI_YLA_ETU
Rev.
Change
Date
Changed
Appr.

Appendix 2: Belt failure mechanisms (Gates PowerGrip, 2014)

Symptoms	Probable cause	Corrective action
Tensile failure	Foreign body in drive	Ensure cover is correctly fitted
	Excessive installation tension	Install at correct tension
	Belt crimped due to improper handling	Observe handling instructions
Tooth shear	Low tension	Install at correct tension
	Seizure of driven part	Eliminate cause
	Misalignment	Correct alignment
Tooth wear	Incorrect tension	Install at correct tension
	Worn pulley(s)	Replace pulley(s)
Hollowed teeth	Extremely low tension	Install at correct tension
	Loss of tension during running	Ensure tensioner screws are tight
Back cracks	High temperature	Eliminate cause
	Low temperature	Eliminate cause
	Back idler is worn out	Replace back idler
Land wear	Excessive tension	Install at correct tension
	Rough pulley(s)	Replace pulley(s)
Oil contamination	Oil leak	Replace faulty oil seals
		
Edge wear	Flange(s) damaged	Replace pulley(s)
	Misalignment	Correct alignment
Noise	High tension	Install at correct tension
	Low tension	Install at correct tension
	Misalignment	Correct alignment
	Flange(s) damaged	Replace pulley(s)